The
ENGINEERS' DIGEST

REVIEW OF ENGINEERING PROGRESS ABROAD

VOL. V. No. 5 MAY, 1944

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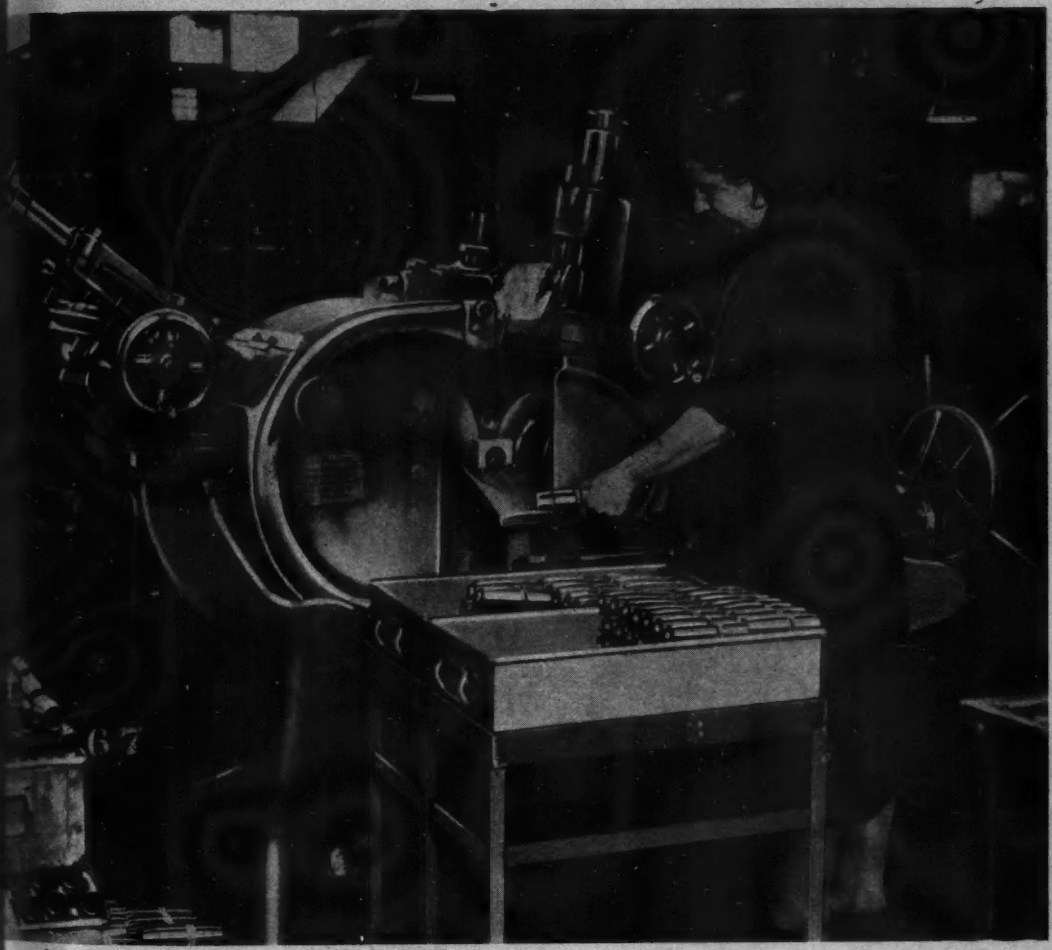

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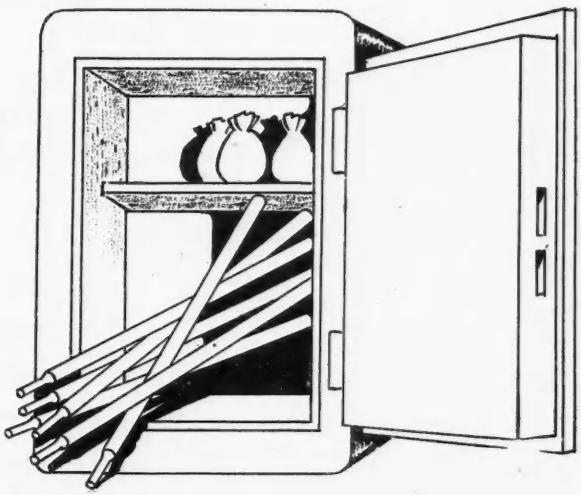

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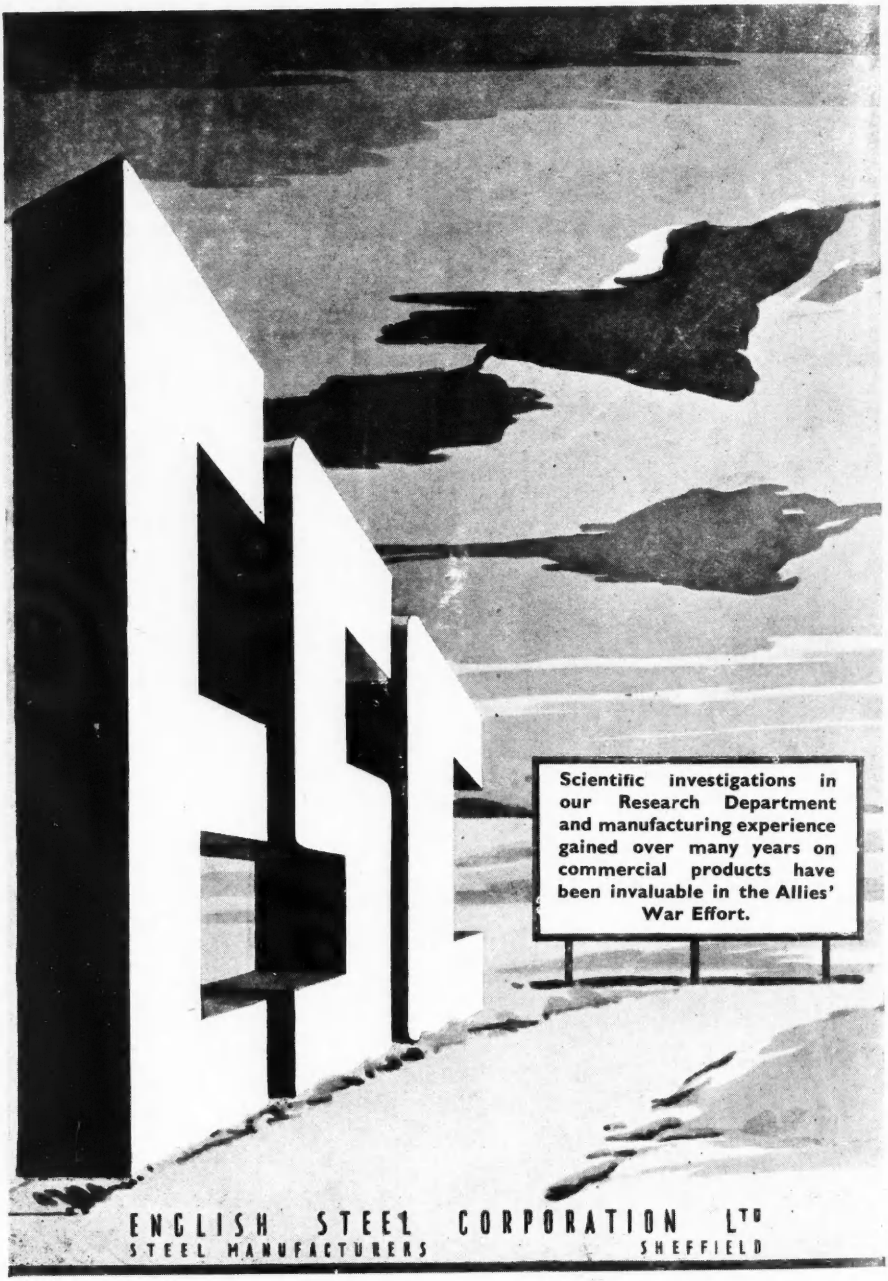
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CALCULATION OF A FUSELAGE NOSE AS A SPACE-FRAMEWORK WITH RIGID NODES.

By Dr. Sc. Techn. El-Sayed El-Schasly. (From *Flugwehr und Technik*, Vol. 5, No. 7, July 1943, pp. 189-194.)

The fuselage nose of multi-engine military aircraft is to provide full view for the crew. To this end a grid of duralumin is covered by plexiglass (Fig. 1). In replacing these trapezoid plates by diagonals of the same shear stiffness as that afforded by the plexiglass, a structure is obtained which closely resembles the Schwedler dome. (A method for calculating the latter has been developed by the present author, and is published as a report of the Institute for Structural Statics of the Swiss Federal Technical College, No. 12, Zurich, 1943.)

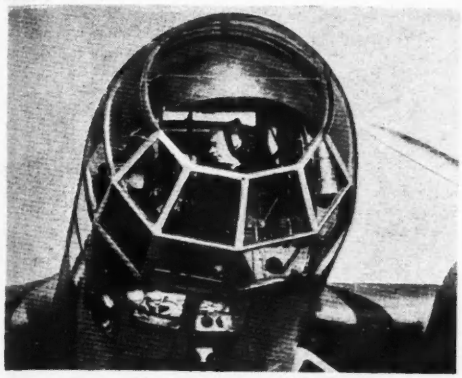


Fig. 1. **The fuselage nose of the Do. 17.**

The trapezoid plate can be replaced by a stepped one, to the edges of which a constant shear flow $q = \tau\,\delta$ can be applied (Fig. 2a). Herein, δ is the thickness of the plexiglass plate, and τ the shear stress. Of the shear load in the plate each node of the framework contains that part which is transmitted by the frame as from middle to middle of the adjoining rods (Fig. 2b). The resulting load on the node is of the magnitude $q\frac{d}{2}$, and in direction of the diagonals. The load of the substituted diagonal is $D = qd$, its elongation $\Delta d = \frac{q\,d^2}{EF}$, where F is the cross-sectional area of the diagonal, E is Young's modulus for the framework (Duralumin $E = 700{,}000$ kg./cm.²). Further, denoting by G the shear modulus of the plexiglass plates ($G = 8{,}000$ kg./cm.²) and, by disregarding the deformation of the frame, the shear strain of the plexiglass plate from Fig. 2c is

$$\gamma = \frac{q}{\delta G} = \frac{\Delta d}{h \cos\beta}$$

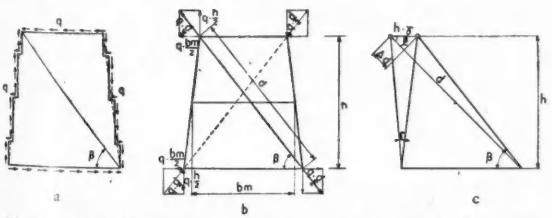


Fig. 2. **Diagonal rod of equal stiffness with the plate it replaces.**

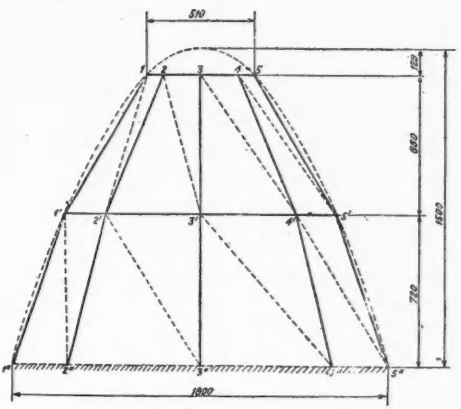
Therefore, the cross-sectional area of the diagonal, equivalent to the stiffness of the plate is

$$F = \frac{q\,d^2}{E\,\Delta d} = \frac{\gamma\,\delta\,G\,d^2}{E\,\Delta d} = \delta\,\frac{G}{E}\,\frac{d^2}{h\cos\beta} \text{ or } F = \delta\,\frac{G}{E}\,\frac{d^3}{h\,b_m}$$

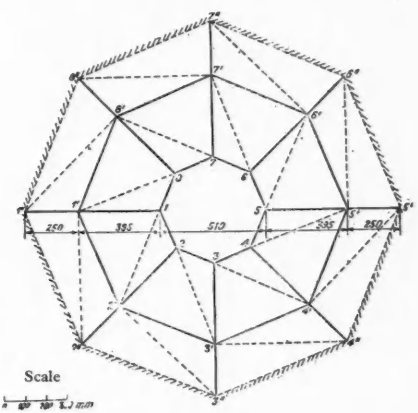
The calculation of the nose is based on two loading cases :—

(a) *Symmetrical Loading*.—This case corresponds to a dive, under the assumption of the nose as a body of revolution, and of axial flow. Assuming a dive speed of 950 km./hr., the dynamic pressure is $q = 3{,}580$ kg./m.² Due to compressibility this increases to $\bar{q} = 4{,}150$ kg./m.² The pressure distribution is obtained from the potential flow theory.

(b) *Asymmetrical Loading*.—This case corresponds to a pull-out from high speed. The dynamic pressure is now 80 per cent of that of Case (a), and by considering the compressibility effect it increases to $\bar{q} = 3{,}228$ kg./m.² The axis of the nose is at three degrees to the direction of airflow.



ELEVATION



PLAN-VIEW

Fig. 3. **Dimensions of fuselage nose.**

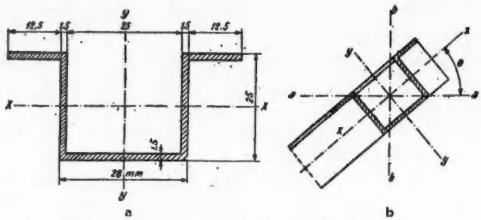


Fig. 4. **Cross-section of rings and rafters.**

The method of calculation is shown by means of an example. As a body of revolution a paraboloid has been chosen. There are eight rafters and two annular members, called the intermediate and end rings respectively (Fig. 3). The dome itself is assumed to be rigidly fixed at its support (i.e., where the nose joins the monocoque structure of the fuselage). The equivalent diagonal is shown dotted.

The cross-section of both rings and rafters is constant throughout, and is of the dimensions given in Fig. 4. The cross-sectional area and moment of inertia of this profile are :—

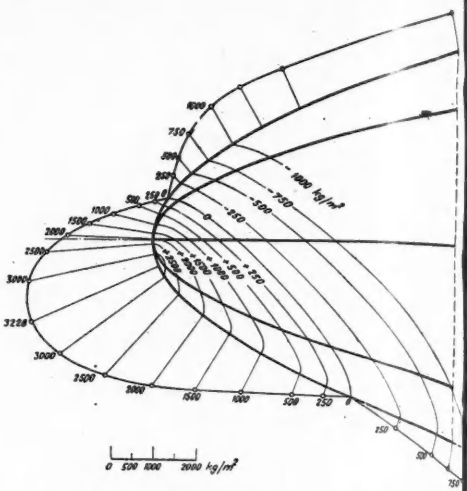
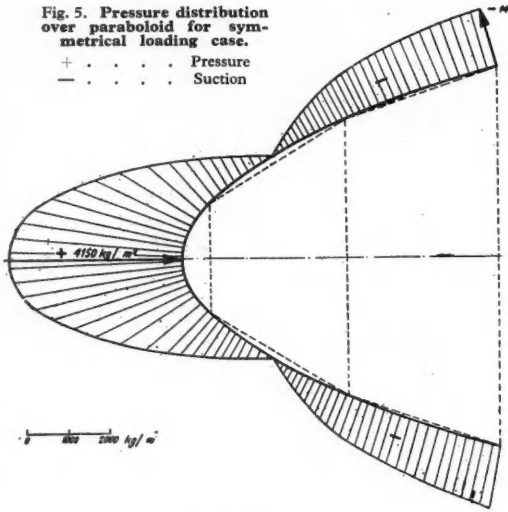


Fig. 6. **Pressure distribution and isobares over paraboloid for assymetrical loading case.**

Fig. 5. **Pressure distribution over paraboloid for symmetrical loading case.**

\+ . . . . . Pressure
— . . . . . Suction



$F_c = 1.500$ cm.²
$I_x = 1.426$ cm.⁴ $= 0.951\ F_c$
$I_y = 3.0986$ cm.⁴ $= 2.066\ F_c$

An increase in these values due to the effective width of the plexiglass plate has not been taken into account.

The dimensions are :—

In the front field $h = 75.4$ $b_m = 34.6$ $d = 83.0$ cm.
In the rear field $h = 75.6$ $b_m = 59.3$ $d = 96.1$ cm.

With a thickness $\delta = 3$ mm. of the plexiglass, the cross-sectional area of front and rear diagonals replacing the plate becomes

$F_f = 2.5 = 0.75$ cm.² $= 0.5\ F_c$ and
$F_r = 2.26 = 0.678$ cm.² $= 0.452\ F_c$

The loading, Cases (a) and (b), are represented in the pressure distributions (Figs. 5 and 6), and result in nodal loads which are given in Figs. 7 and 8.

Calculation of Forces and Stresses of Rods of the Statically Determinate Basic System.

To be able to superimpose any arbitrary loading case, the calculation is done for single loads $P = 1$ and $W = 1$ at one node of the end ring and one node of the intermediate ring, as well as for $H = 1$ at one node of the end ring only. The loads P, W and H act vertically, horizontally radial and horizontally tangential respectively to the plane of the ring.

The internal rod forces S_o of the statically determinate dome (ball-jointed) for the asymmetrical loading is shown in Fig. 9, the resulting stresses $\sigma = \frac{S_o}{F_c}$ are shown in Fig. 10. For the symmetrical loading case these values are as follows :—

Rod	1—2	1—1′	1′—2′	1′—1″
Force (kg.)	—319	—378	+541	—250
Stress (kg./cm.²)	—213	—252	+301	—167

Calculation of the Dome with Flexurally Rigid Rafters and Ball-jointed Rings and Diagonals.

The dome is considered as the basic system into which the rafter moments X_1' and X_1'' are introduced as unknowns. For the octagonal dome the single

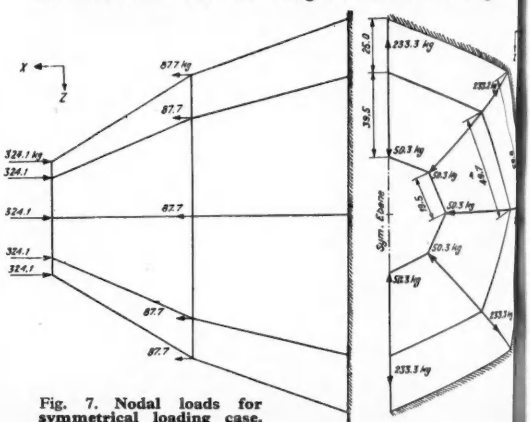


Fig. 7. **Nodal loads for symmetrical loading case.**

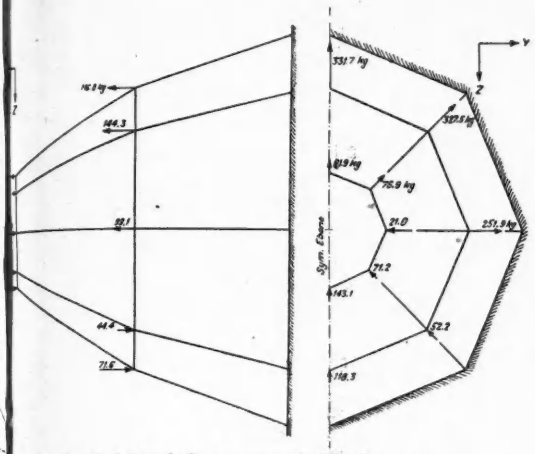


Fig. 8. **Nodal loads for asymmetrical loading case.**

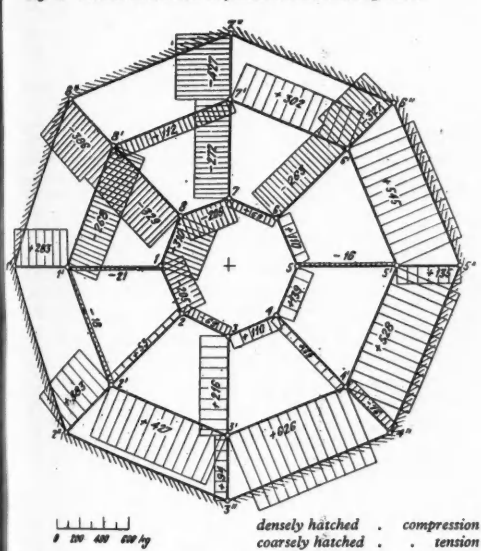


Fig. 9. **Rod forces (kg.) of dome with ball-jointed nodes for asymmetrical loading case.**

nodal load $P_1 = 4$ tons can be regrouped into four loads whereby the number of the equations of elasticity is considerably reduced. The rod forces $S(r)$ with flexurally rigid rafters can be obtained from those found in the previous chapter (S_o) :—

$$S(r) = S_o + S_o' X' + S_o'' : X''$$

Herein S_o' and S_o'' refer to $X' = 1$ kg./cm. and $X'' = 1$ kg./cm. respectively.

Calculation of the Dome with Flexurally Rigid Rafters and End-Ring. The rest of the Rods are Ball-jointed.

The system described in the previous chapter serves as the statically indeterminate basic system.* A general term $a_{1k}(n)$ gets the following form :—

$$a_{1k}(n) = \Sigma S_1^o \cdot S_k^n \cdot L/EF + \int M_1^o \cdot M_k^n \cdot ds/EI$$

* See the publication of the author previously referred to.

The final rod forces are shown in Fig. 11, for the asymmetrical loading case. The stresses for this loading case are shown in Figs. 12 and 13, for the inner and outer fiber of the rod cross-section respectively.

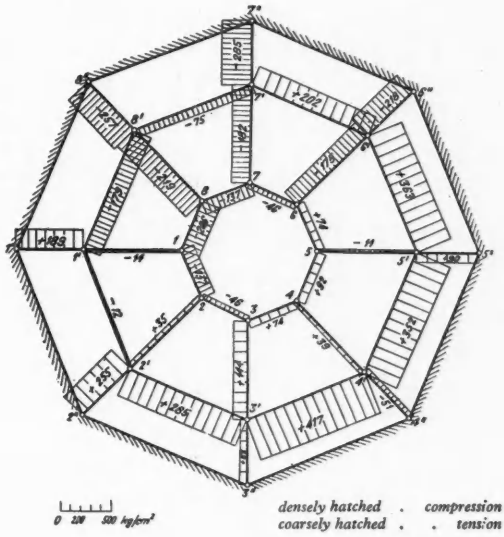


Fig. 10. **Stresses (kg./sq. cm.) in dome with ball-jointed nodes for asymmetrical loading case.**

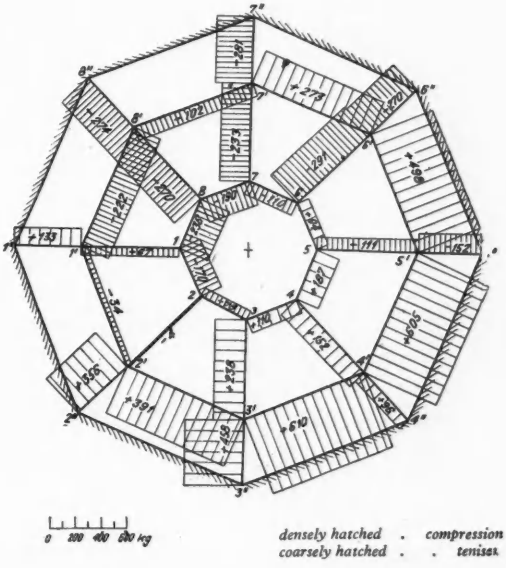


Fig. 11. **Rod forces (kg.) of dome with rigid nodes for asymmetrical loading case.**

For the symmetric loading case the forces and stresses are as follows :—

Rod ..	1—2	1—1′	1′—2′	1′—1″	Force (kg.) ..	—318	—377	+449	—249
Stresses (kg./cm.²):									
Maximum ..			—175	—228	+300	—142			
Minimum ..			—249	—275	..	—189			

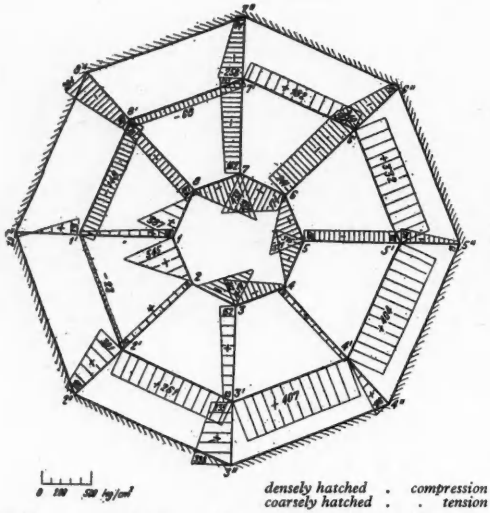


Fig. 12. **Stresses (kg./sq. cm.) in outer fibres of rods of dome with flexurally rigid rafter and end ring for asymmetrical loading case.**

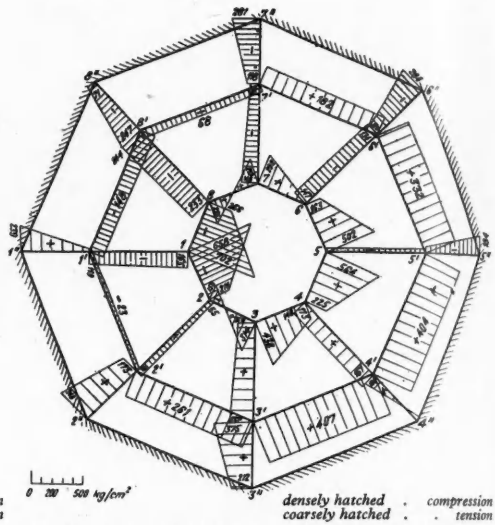


Fig. 13. **Stresses (kg./sq. cm.) in inner fibres of rods of dome with flexurally rigid rafter and end ring for asymmetrical loading case.**

SWISS GLIDER CONSTRUCTIONS.

By TH. HEIMGARTNER, Zurich. (From *Swiss Technics*, Vol. 22, No. 4, November-December, 1943, pp. 21-24).

WHEN Eric Nessler, flying a Swiss designed glider, made the 38-hour record for France (this record has been beaten since), the whole of the flying world, far beyond the boundaries of France, immediately became interested in the S-18-Glider constructed by J. Spalinger.

To-day international flying circles announce gliding as the "aviation sport of the future." The flying performances of the latest Swiss gliders are in no way inferior to those of the corresponding foreign types. And with regard to their manœuvrability they are clearly superior to almost all foreign models of the same category. Owing to the fact that Swiss machines have very often to be dismantled when they are to be used on high alpine flying-fields (mountain railway transport), special attention has been given to ease of assembly and dismantling. In order to be able to land on very small fields, especially in mountainous countries, all the more recent machines are equipped with dive or landing brakes, which have proved to be very effective.

In the case of the one-seater gliders, the following models come into consideration :

Type : *S*-18-*III* is a medium size glider (wing-span 14.2 metres) constructed by J. Spalinger. It has very desirable flying qualities, not the least of which is its careful aerodynamic design. It is outstandingly adapted for thermic flights combined with a very low minimum flight velocity.

Type : *Spyr IV* is designed by A. Hug specially for high-speed distance flights. Military restrictions on flying routes prevented this aircraft from proving its qualities. Nevertheless M. Godinat attained an altitude increase from starting level, of over 3000 metres, which proves that, at least in altitude flights, it is a very

Fig. 1
Two-seater glider, "Spyr V."
Designer: A. Hug.

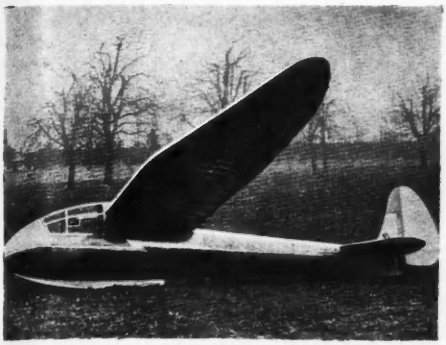


Fig. 2. **One-seater glider S-18-III.** Designer: J. Spalinger

efficient machine. Up to now there are only two machines of this type, which has a wing-span of 16.4 metres, and weighs, empty, 180 kg. Long distance records can with certainty be expected from this high-quality glider.

Type: *Moswey III* is a modification of the serially-produced Moswey II, which was the winner in the National Gliding Championship Competition of 1942. This high-quality glider, designed by J. Müller, has a very great range of speed and a striking manœuvrability. This type, which has proved itself especially adaptable for high alpine flights, has a wing-span of 14 metres and weighs, empty, 130 kg. It is unusually sturdy. The machine is also æsthetically pleasing; it is at present manufactured in series. Sweden inaugurated the export of these machines by ordering one of the first Moswey series.

The Spyr-V is a two-seater designed by A. Hug. It replaces the "Geier" two-seater previously designed by Hug, and has a wing-span of 18.4 metres, weighing, when empty, 265 kg. The test flights of this type, which unmistakably exhibits the characteristics of the traditional Spyr construction (slender wing tips), gave very satisfying results. Entirely new and very original are the dive-brakes of the Spyr V. The adjacent pilot seats are arranged slightly obliquely to keep the width of the fuselage within acceptable limits. The prototype is used at present for meteorological flights.

The two seater types designed by J. Spalinger exhibit the following characteristics:

Type: *S-21-I*: This is of normal construction in wood with oval fuselage and braced-gull-wings. The passenger cabin, which, like all Swiss two-seaters, is equipped with dual-control, is placed under the wing.

Fig. 3. **One-seater glider "Spyr IV."** Designer: A. Hug.

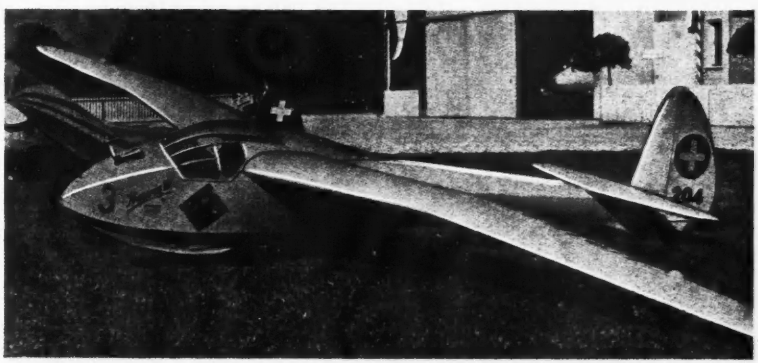


Fig. 4. **One-seater glider "Moswey II."** Designer: G. Müller.

Fig. 5. **Two-seater glider, "S-21-I" (wooden fuselage).** Designer: J. Spalinger.

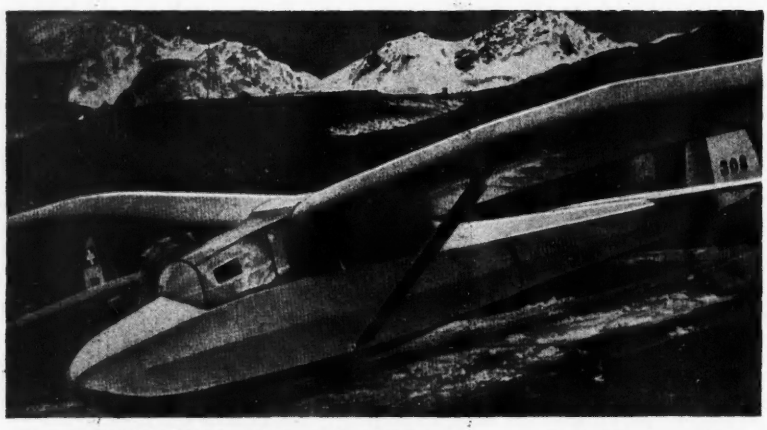


Fig. 6.
Two-seater glider, "S-21-II," (Tubular steel fuselage).

Type: *S-21-II* (Fig. 6): This type is similar to its forerunner of the S-21 series. The fuselage is, however, built of steel tubing. An innovation is the arrangement of the seats, which lie slightly obliquely in front of the middle cross-section. Owing to this arrangement, the fuselage is very broad, though this has not caused any appreciable decrease in efficiency or manœuvrability. On the contrary, actual tests have shown that, during high alpine flights, this machine is able to maintain, on the average, the same altitudes as the one-seater gliders. Apart from the clear vision for both passengers, and the good communication possibilities, it has the further advantage that the dash-board instruments need not be duplicated, as they are visible for both pilots.

Type: *S-25*: Since the prototype is still in the development stage, nothing definite can be stated as yet concerning the flying qualities of this very promising type. However, this two-seater glider presents an ideal solution of the questions of space, equipment, visibility and practical seating-arrangement, a solution which will probably not be easily improved upon.

Technical data and flight characteristics for the most common types of Swiss gliders.

One-Seaters.

Glider Type:	*S—18—III*	*Spyr—IV*	*Moswey III*
Wing-spread ..	14.3 m.	16.4 m.	14 m.
Length	6.3 m.	6.5 m.	6 m.
Weight, empty ..	155 kg.	180 kg.	130 kg.
Carrying capacity ..	85 kg.	80 kg.	95 kg.
Flying weight ..	240 kg.	260 kg.	230 kg.
Gliding ratio ..	1 : 21	1 : 30	1 : 25
Span loading ..	16.8 kg./m.	19 kg./m.	18.5 kg./m.
Normal flying speed	60 km./hr.	63 km./hr.	68.4 km./hr.
Speed of descent ..	76 cm./sec.	69 cm./sec.	76 cm./sec.

Two-Seaters.

Glider Type:	*Spyr—V*	*S—21—I*	*S—21—II*	*S—25*
Wing-spread ..	18.4 m.	17.3 m.	17.5 m.	17.7 m.
Length	7.8 m.	7.6 m.	7.8 m.	8.4 m.
Weight, empty ..	265 kg.	219 kg.	220 kg.	250 kg.
Carrying capacity	160 kg.	160 kg.	160 kg.	160 kg.
Flying weight ..	425 kg.	379 kg.	380 kg.	410 kg.
Gliding ratio ..	1 : 25	1 : 21	1 : 20	1 : 22
Span loading ..	20 kg./m.	18.8 kg./m.	18.8 kg./m.	20 kg./m.
Normal flying speed	65 km./hr.	62 km./hr.	62 km./hr.	65 km./hr
Speed of descent	76cm./sec.	78cm./sec.	78 cm./sec.	80cm./sec.

PRESSURE CONTROL OF STEAM MAINS SUPPLIED FROM EXTRACTION TURBINES.

By DIPL.-ING. W. HERMANN. (From *Die Wärme*, Vol. 66, No. 14, April-May, 1943, pp. 120-121.)

IN order to achieve highest steam plant economy, extraction steam is being increasingly used for feed heaters, evaporators, steam transformers, and de-aerators—and this in addition to its employment in manufacturing processes carried out in industrial plants.

As the pressure prevailing at a bleeder point depends on the turbine load (as shown in Fig. 1), extraction steam line pressure control will be found necessary wherever extraction steam is furnished jointly by several turbines. Generally, the extraction point is so chosen that the required line pressure equals that of the bleeder steam at one-half the normal turbine load. When this load point is exceeded, throttling of the bleeder steam will therefore be required; while at smaller loads, steam flow reversal at the extraction point must be precluded by means of a non-return valve, or by automatic closure of the reducing valve in the bleeder line.

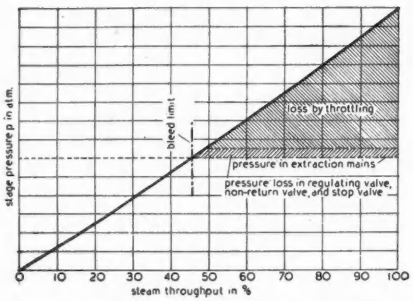


Fig. 1. Pressure characteristic at extraction point.

The type of control hitherto used is shown in Fig. 2. Here each turbine is provided with an automatic regulator which is actuated by the pressure of the extraction steam mains. If the latter pressure decreases, the controlling organ of the regulator is actuated, and this in turn adjusts the position of the reducing valve. This arrangement, however, suffers from the disadvantage that, with the simultaneous extraction from two turbines, the turbine operating at higher load—with correspondingly higher pressure prevailing at the extraction point—will be called upon by the controller to supply more extraction steam than the other turbine operating at lower load. As this is due to the fact that for the same opening of the reducing valve, the amount of steam actually passed depends on the upstream pressure—that is, on the pressure at the extraction

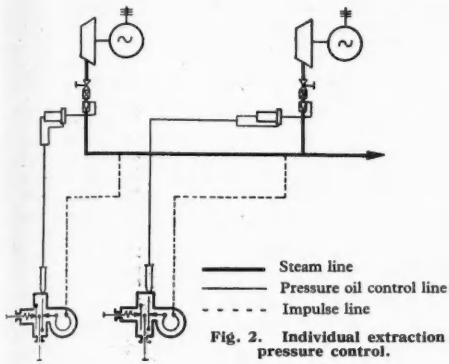
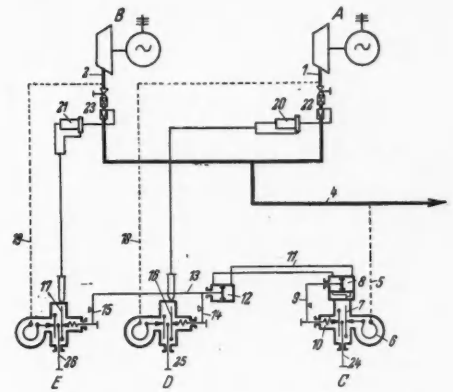


Fig. 2. Individual extraction pressure control.

point—the control pressure of the controller can be correspondingly adjusted by hand in order to restore the situation. This deficiency of the control action is eliminated in the patented* control arrangement shown in Fig. 3, which ensures that extraction steam is always taken from the turbines, the bleeder point pressure of which is closest to the required extraction pressure. In this way, throttling losses are reduced to a minimum.

Where extraction steam is provided by more than one turbine, the withdrawal of extraction steam is made to take place in the order given by the respective deviations of the bleeder point pressures from that of the extraction mains, with a corresponding change in sequence with changing load conditions. This is effected by the control arrangement shown in Fig. 3, in which the extraction mains 4 is connected to bleeder points 1 and 2 of turbines A and B respectively. The master controller C receives its control impulse from the pressure prevailing in line 4 by means of the connection 5. The pressure sensitive organ of the master controller is a Bourdon tube 6, which acts upon the relay 7, which in turn actuates the power cylinder 8. The latter is connected by the rigid lever system 9 to the pressure setting device 10. The movement of the piston 8 is transmitted to the slave piston 12, the piston rod of which operates the actuating devices 16 and 17

* German Patent, 719,189.



Steam line ——— Pressure oil control line
- - - - - Impulse line

Fig. 3. Control gear for high-efficiency extraction control.

of the controllers D and E respectively by means of a link and lever system.

These control forces, which tend to establish equal extraction steam delivery by each turbine, are counteracted by opposing control forces 18 and 19 respectively, which are derived from the extraction point pressure of the turbine concerned. The resultant of the two opposing impulses is made to act upon the power cylinders (20 and 21 respectively), which in turn operate the reducing valves (22 and 23 respectively).

Since the magnitude of the counter-impulse increases with the extraction point pressure, and, consequently, with the turbine load, the extraction valve of the turbine carrying the lowest load is made to open wider than the others, thus causing this machine to deliver the largest amount of extraction steam. In order to avoid excessive loss by throttling, the extraction is preferably limited to a fixed load range. In cases where very low loads occur at times (low night or week-end loads) the turbine is preferably equipped with two bleeder stages, provision being made for automatic switching over from one extraction point to the other where called for by the prevailing load conditions.

PLASTIC PLYWOODS IN AIRCRAFT CONSTRUCTION.

By R. D. Hiscocks, Engineer in Charge of the Structural Laboratory, Aeronautical Division, National Research Council, Ottawa, Ont. (From *The Engineering Journal*, Vol. 27, No. 1, January, 1944, pp. 4-8).

WOOD AS STRUCTURAL MATERIAL.

Wood is admittedly far from the isotropic ideal dear to the elastician; its elastic properties vary with moisture content, specific gravity and even from the heartwood to the tapwood of the same tree. Thus, wood may appear a vague proposition for an engineering material, but if the external loading conditions of an airframe are examined, it is apparent that the only logical assessment should be on a statistical basis. Therefore, there is no inconsistency in employing a material the strength properties of which are not uniquely determined. Besides, stiffness rather than ultimate strength governs the design. In elastic stability problems the ratio $\frac{E}{\rho}$, or $\frac{E}{\rho_2}$, or even $\frac{E}{\rho_3}$ (where E = Young's modulus, ρ = density) is the figure of merit, and under such conditions an expanded material, such as wood, is supreme. The problem of load diffusion is frequently less difficult with plywood than with metal structures; with longerons, for example, it is possible to taper the cross sectional area and to graduate the elastic properties by varying the construction so that the glue joint between the longeron and the skin is uniformly stressed. Finally, with the insistence of the aerodynamicist on smooth surfaces, it is not difficult to see that present metal construction is considerably handicapped.

TESTS TO OBTAIN BASIC DATA.

Of the plywood cylinders tested in the Structural Laboratory some were hot-press moulded with a rubber bag in the autoclave, others were formed with two-ply veneer and cold press resin on a wrapping machine. The grain of the wood was in all cases 45 deg. to a generator, with inner and outer plies parallel and the grain of the core plies at right angles. The radius R of the cylinders was between $R=3$ to $R=12$ inches.

These tests have shown that the compressive buck-

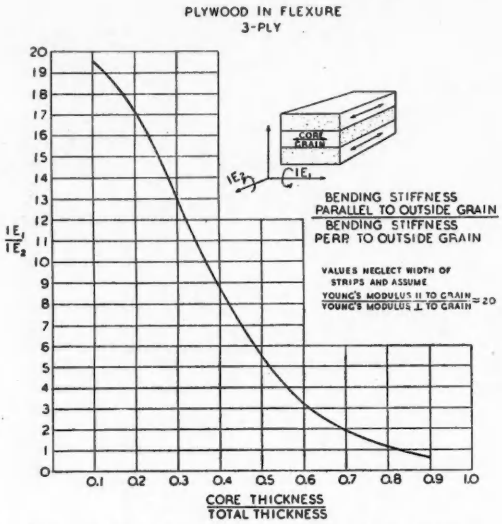

PLYWOOD IN FLEXURE
3-PLY
CORE GRAIN
BENDING STIFFNESS PARALLEL TO OUTSIDE GRAIN
BENDING STIFFNESS PERP TO OUTSIDE GRAIN
VALUES NEGLECT WIDTH OF STRIPS AND ASSUME
YOUNG'S MODULUS // TO GRAIN / YOUNG'S MODULUS ⊥ TO GRAIN = 20
CORE THICKNESS / TOTAL THICKNESS

Fig. 1. **Effect of core thickness on plywood in flexure.**

ling stress σ_b can be expressed in the same form as with metal structures, $\sigma_b = KE\frac{t}{R}$, and the experimental constant $K=0.3$ is also the same, being in these tests on the conservative side. The modulus of elasticity E is to be taken parallel to the direction of the load. t is the wall thickness.

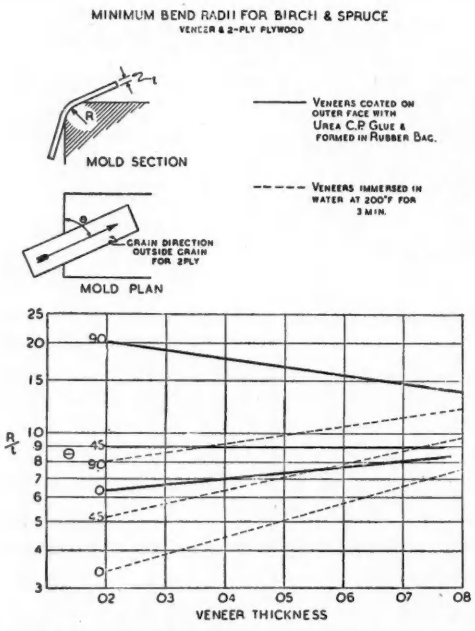

MINIMUM BEND RADII FOR BIRCH & SPRUCE
VENEER & 2-PLY PLYWOOD
MOLD SECTION
VENEERS COATED ON OUTER FACE WITH UREA C.P GLUE & FORMED IN RUBBER BAG.
VENEERS IMMERSED IN WATER AT 200°F FOR 3 MIN.
GRAIN DIRECTION OUTSIDE GRAIN FOR 2PLY
MOLD PLAN
VENEER THICKNESS

Fig. 2. **Minimum bend radii for veneer and two-ply plywood.**

The low bearing strength of wood perpendicular to the grain can be avoided by the use of plywood. With concentrated loads the need arises for an intermediate material having high bearing strength such as phenolic fabric base laminates of the order of 30,000 lb. per sq. in.

BONDING AND GLUING.

Phenolics usually will bond to wood effectively only with hot press resins. The most satisfactory expedient is to first glue a thin layer of veneer to the laminate with hot press adhesive and then to bond the wood face to the remainder of the structure with a cold setting adhesive. The same considerations apply in bonding metal to wood. As wood has no elastic limit in tension, care should be taken when diffusing a concentrated load into a shell structure. Highly localised tensile stresses may cause failure even at a moderate average stress because the material cannot yield plasticity and distribute the local stresses.

High pressures are not required in gluing (one lb. per sq. in. feasible) if the viscosity of the glue is not excessive. In gluing it is of the utmost importance to match the moisture content of the pieces to be glued together. Wood is a hygroscopic material and shrinking and swelling takes place with changes in the moisture content of the air. Most of the dimensional change occurs in a direction perpendicular to the grain and, for a stable material three plies of veneer are the minimum.

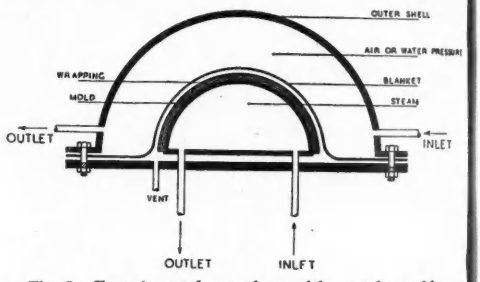


Fig. 3. **Experimental autoclave with metal mould.**

Unfortunately, conventional three-ply is composed of veneers of equal thickness and the bending stiffness parallel to the face grain is about 12 times the stiffness at right angles. In general it would be preferable to employ a construction which gives a comparable stiffness in both directions. Fig. 1 shows how this can be done by increasing the core thickness over the thickness of the two face plies.

BENDING, COOKING AND MOULDING PLYWOOD.

The safe bend radius in design of shell structure is shown in Fig. 2.

To accelerate the drying of coldsetting resin adhesives infra-red lamps are employed. Charts are available showing the urea cold press drying time plotted against temperature. The time is very considerable at temperatures which prevail in workshops. (Several days). If however, a thick pack of veneers is cooked under a 260 watt infra-red lamp, the total time required would be 60 minutes for a glue to set. This value is obtained for a thickness to furthest glue line of $\frac{5}{8}$ inch, and the temperature measured was 120 deg. F.

The bag moulding art has received a good deal of attention in the literature lately and requires no description. We believe that a metal mould gives satisfactory results. Fig. 3 shows a section through such a mould. Compressed air can provide the required pressure independently of the temperature of the heating medium which is preferably steam.

THE HEAT PUMP AS REFRIGERATING AND HEATING MACHINE.

By A. MELDAHL. (From *Brown Boveri Review*, Vol. 30, Nos. 5/6, May to June, 1943, pp. 75-85).

FUNDAMENTALLY, there is no difference between refrigerating and heating machines; both are heat pumps. The refrigerator is well-known, but the heating machine is little known because of the many possibilities of heating without a machine by means of fuel or electricity. Heating machines possess many advantages and can supply a much greater quantity of heat with the same consumption of energy, as would be available by direct conversion of the same energy into heat, and in many cases more than compensates the higher price of electrical energy compared with that of fuel.

A heating machine may be used for warming a building in winter and for cooling in summer, a dual provision which is an economical justification for the machine. To judge the possibilities of the machine, a knowledge of certain thermo-dynamics as well as experience in the subject, is indispensable and the clearest view of the principle employed is obtained by comparing it with a heat engine which converts heat into mechanical work. Heat engines are of two main classes: steam engines and gas engines. In the steam engine (Fig. 1) a feed pump delivers water into the boiler in which heat causes the water to evaporate, whereby its volume increases. The steam, in expanding to a lower pressure in the engine, does work, after which it is condensed and gives up its latent heat; the condensate then returns to the feed pump.

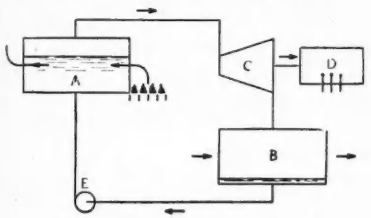


Fig. 1. Diagram of a steam engine.
(A) Heated boiler; (B) Cooled condenser; (C) Steam engine; (D) Generator; (E) Feed pump.

This fundamental principle is evident: the engine can only operate when the boiler pressure is higher than that of the condenser, that is, when the boiler temperature is higher than that of the condenser, and in order to feed the water into the boiler, the steam must be condensed so that it can no longer be considered for the production of mechanical energy.

In the gas engine, the gas is compressed and heated, expanded and finally cooled. During expansion of the hot gas more energy is produced than is needed for compression of the cold gas and the difference is the useful output. The main difference compared with the steam engine is that the feed pump takes less than 1 per cent. of the output of the steam engine, and in the gas engine the work of compression is more than 50 per cent. of the expansion work.

HEAT PUMPS.

The heat pump is nothing else than a reversed heat engine. In place of the rejection of heat at the lower temperature, there is the absorption of heat, the heat engine *develops* power, whereas the heat pump *absorbs* power. Such a machine can, therefore, absorb heat at a lower temperature and give it up again at a higher temperature—the reverse process of that observed in nature. It raises heat to a higher temperature level—hence the name "heat pump."

Heat pumps can be used as refrigerating or as heating machines. The refrigerator absorbs heat at a low temperature, but the heating machine gives up heat; the useful output is the heat given up at the higher temperature from that of the lower temperature which may be derived from the surroundings, the air, a lake, or some source of exhaust heat. The ratio of the heat given out to the heat equivalent of the energy absorbed is known as the coefficient of performance, and indicates how much more heat is supplied by the heating machine than by direct electrical heating for the same consumption of energy.

THE STEAM HEAT-PUMP.

Fig. 2 shows a steam heat-pump, the reverse of the steam engine (Fig. 1). The working medium is

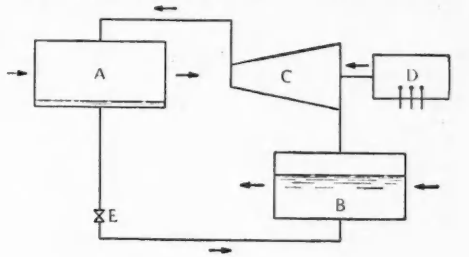


Fig. 2. Diagram of heat pump.
(A) Cooled condenser; (B) Evaporator heated with waste heat or heat from the surroundings; (C) Compressor; (D) Motor; (E) Reducing valve.

evaporated at low pressure in an *evaporator* which replaces the condenser. The water circulated supplies the necessary latent heat of evaporation and the low-pressure steam produced is absorbed and compressed by the compressor which absorbs power. Instead of a generator there is a motor and the compressed steam is condensed in a *condenser* which takes the place of the former boiler. The latent heat of evaporation is given up to the cooling water passing through the condenser which is thereby heated. Whereas, in the steam engine the combustion gases are *hotter* than the cooling water, the "heating water" of the heat pump is *colder* than the cooling water.

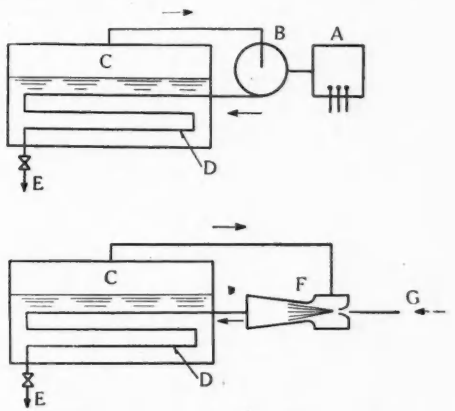


Fig. 3. Diagram of a thermo-compressor.
(A) Motor; (B) Compressor; (C) Cooker; (D) Heating coil; (E) Condensate outlet; (F) Steam heat compressor; (G) Live steam.

A simple form of steam heat-pump is shown in Fig. 3, when a concentration process is operated by means of a heating machine and the steam produced is itself used as a working medium. In this case the heating machine consists of what is known as a *thermo-compressor* which makes use of the latent heat of evaporation of the steam and which would otherwise be lost.

The working medium employed depends on the conditions. At low temperatures (CO_2), (NH_3), and (SO_2) have relatively high vapour pressure, and low specific volume; they are, therefore, suitable for reciprocating compressors.

THE GAS HEAT-PUMP.

The gas heat-pump is also a reversed gas engine. The working medium is compressed adiabatically and thereby heated; the heated gas flows through a heat exchanger, where it gives up its heat, after which it is expanded and cooled so that it may take up heat at a lower temperature in a secondheat exchanger. The requirement of an expansion machine makes the gas heat-pump more complicated than the steam heat-pump, nevertheless, for certain purposes where low temperatures must be attained, as in high altitude test plants for aircraft engines, gas heat-pumps are of advantage.

THERMO-DYNAMIC FUNDAMENTALS—HEAT PUMPS.

As already mentioned the heat pump is nothing else than a reversed heat engine and the working process of the steam heat-pump (Fig. 4) would, for instance, be as follows. Boiling water (point 1) is converted to steam by heat supplied (5), the steam is adiabatically compressed (4), then condensed (3) and the condensate is again expanded to the initial pressure.

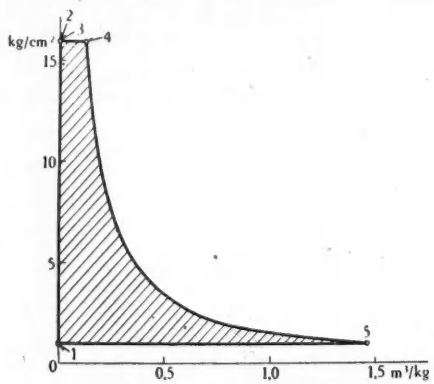


Fig. 4. **p-v diagram of a steam heat pump.**
(p) Pressure; (v) Specific volume.

The heat quantities may be read off from the entropy diagram (Fig. 5). The heat supplied is represented by the surface below the line 1, 5; the quantity of heat given off by the surface below the line 4, 3, 2, 1, and the power by the surface 1, 5, 4, 3, 2, 1. By this means a certain amount of heat is taken in at the lower temperature and given out at a higher temperature. The surface below the entropy diagram is the heat absorbed and the two together represent the heat which the heat pump can give out. It is customary to express the performance of the heat pump as the ratio of useful heat to the power absorbed; this is called the coefficient of performance and may also be expressed in heat units. The theoretical max. value of the coefficient of performance may be deduced from Carnot's law, depending, however, upon which quantity is regarded as the useful heat, for a heat pump may operate either as a refrigerator or as a heating machine.

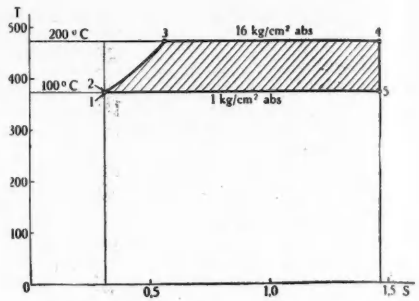


Fig. 5. **T-S diagram of a steam heat pump.**
(T) Absolute temperature; (S) Entropy.

In a refrigerator the maximum ratio of useful output to power absorbed is

$$\epsilon_{th\,K} = \frac{T_2}{T_1 - T_2}$$

The minus sign comes from the fact that the useful output is a heat quantity taken in. T_1 is the upper temperature limit which, in refrigerator practice, is approximately 300 deg. C. abs. The coefficient of performance is fixed by the choice of the lower temperature T_2, and the entropy diagram shows how important it is not to make this temperature a single degree lower than absolutely necessary. At very low temperatures such as the liquefaction of air, or of hydrogen (20 deg. abs). or Helium (4 deg. abs.) enormous amounts of energy must be supplied to produce very small outputs of cold.

In the heating machine, on the other hand, the maximum ratio according to Carnot's law is

$$\epsilon_{th\,H} + \frac{T_1}{T_1 - T_2}$$

The available electrical or mechanical energy could also be directly converted into heat and used for heating purposes. Values of the coefficient of performance depend on the operating conditions and in the case of heating machines which make the heat of the surroundings available for room heating, the lower temperature is fixed at about 300 deg. abs. (Fig. 6). There is, moreover, another interesting field of application for heating machines, both temperatures may be raised and the useful output steadily increases (Fig. 7).

According to Carnot, the ideal coefficient of performance ϵ_{th} cannot be achieved in practice. If ϵ is the actual coefficient of performance, then the efficiency of the heat pump is $\eta = \epsilon / \epsilon_{th}$, which efficiency differs according to whether the heat pump works as a refrigerator or as a heating machine, and is affected mainly by the temperature drop in the heat exchangers, the working process and nature of working substance, and efficiency of the machines used. In the following, therefore, only the effect of the different conditions is shown in quite a general way.

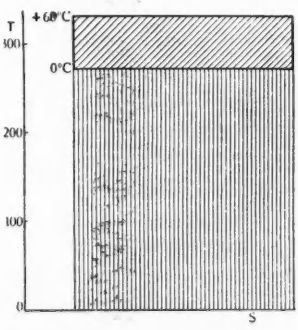


Fig. 6.
Carnot diagram of a heating machine.
////// Theoretically necessary energy consumption.
|||₁|| Heat recovered from the surroundings. The lower the upper temperature, the smaller the energy consumption.

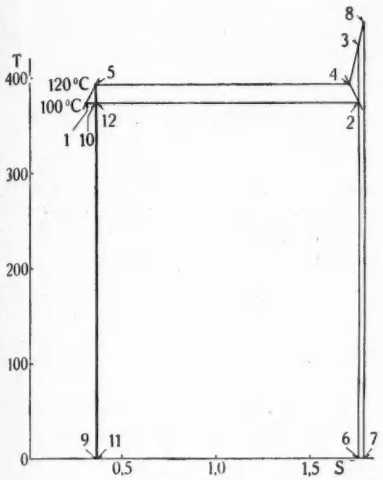


Fig. 7. **Entropy diagram of a thermo-compressor.**
Evaporator temperature 100 deg. C. Condenser temperature 120 deg. C.
1, 2, 3, 4, 5, 1 = Theoretical compression work.
2, 6, 7, 8, 3, 2 = Additional work due to compressor losses.
2, 6, 9, 10, 2 = Theoretically recoverable latent heat of evaporation.
10, 9, 11, 12, 10 = Losses due to throttling and subsequent evaporation of the condensate.

(*a*) *Temperature Drop in the Heat Exchangers.*

Let T_o = lower temperature.
T_1 = upper temperature.
δT_o = temperature drop in the evaporator.
δT_1 = temperature drop in the condenser.
$\Delta T = T_1 - T_o$ = temperature difference to be overcome.

The theoretical coefficients of performance according to Carnot are :

$$\epsilon_{th\,K} = -\frac{T_o}{\Delta T} \text{ for the refrigerator.}$$

$$\epsilon_{th\,K} = +\frac{T_1}{\Delta T} \text{ for the heating machine.}$$

The temperature drop at the heat exchangers causes the temperatures to be displaced ; the heat pump must in reality work under the following conditions :—

$T_o - \delta T_o$ = lower temperature
$T_1 + \delta T_1$ = upper temperature
$\Delta T + \delta T_o + \delta T_1$ = temperature difference to be overcome and the corresponding coefficients of performance of an ideal machine are

$$\epsilon_{1\,K} = -\frac{T_o - \delta T_o}{\Delta T + \delta T_o + \delta T_1} \text{ for the refrigerator}$$

$$\epsilon_{1\,H} = +\frac{T_1 + \delta T_1}{\Delta T + \delta T_o + \delta T_1} \text{ for the heating machine,}$$

from which it follows :

$$\frac{\epsilon_{1\,K}}{\epsilon_{th\,K}} = \frac{\Delta T}{\Delta T + \delta T_o + \delta T_1}\left(1 - \frac{\delta T_o}{T_o}\right) \text{ for the refrigerator}$$

$$\frac{\epsilon_{1\,H}}{\epsilon_{th\,H}} = \frac{\Delta T}{\Delta T + \delta T_o + \delta T_1}\left(1 + \frac{\delta T_1}{T_1}\right) \text{ for the heating machine,}$$

A temperature drop δT_o at the lower temperature is therefore always more harmful than a temperature drop δT_1 of the same magnitude at the higher temperature.

(*b*) *Effects of Process and Working Medium.*

All reversible processes are equivalent independently of the working medium, but when the cycle becomes irreversible the medium effects the coefficient of performance, depending on the process chosen. In the case of steam heat-pumps the process is imperfect in two respects. During compression the steam is heated to a higher temperature than that at which the heat is given up and the work of compression is increased uselessly, and by throttling of the condensate a large amount of expansion work is lost.

The condensate is, wherever possible, cooled below the boiling point and is expanded or throttled in stages. The simplified process, therefore, gives a deformed and exaggerated picture of the effect of different working media.

Let
L_v = the ideal work of compression, that is adiabatic to the higher temperature, then isothermal to the condenser pressure,
L_E = the ideal work of expansion of the boiling condensate,
Q_o = ideal amount of heat taken up,
L_{1v} = the theoretical work of compression during adiabatic compression to condenser pressure,
$a_v = L_v/L_{1v} < 1$,
then for the ideal process

$$\epsilon_{1\,K} = -\frac{Q_o/A}{L_v - L_E} \text{ for the refrigerator}$$

$$\epsilon_{1H} = +\frac{Q_o/A}{L_v - L_E} + 1 \text{ for the heating machine}$$

and for the chosen comparison process

$$\epsilon_{2K} = -\frac{Q_o/A - L_E}{L} \text{ for the refrigerator}$$

$$\epsilon_{2H} = \frac{Q_o/A - L_E}{L_{1v}} + 1 \text{ for the heating machine}$$

and upon rearrangement

$$\epsilon_{2K} = \alpha_v \left\{ \epsilon_{1K} - (\epsilon_{1K} - 1)\frac{L_E}{L_v} \right\}$$

$$\epsilon_{2H} = \alpha_v \left\{ (\epsilon_{1H} - 1) - \epsilon_{1H}\frac{L_E}{L_v} \right\} + 1$$

Therefore those working media are advantageous for which the ratio L_E/L_v is small, that is, having a high latent heat of evaporation and a low specific heat in the liquid state. In the steam state a high specific heat is desirable in order that α_v may be as large as possible. These conditions are partially contradictory ; in general, working media with a high molecular weight are to be preferred.

(*c*) *Effect of the Efficiencies of the Compression and Expansion.*

Let
L_{1v} = theoretical work of compression.
L_{1E} = theoretical work of expansion of the chosen process.
η_v = adiabatic efficiency of the compressor.
η_E = adiabatic efficiency of the expansion machine.
Then the coefficient of performance becomes

$$\epsilon_{2K} = -\frac{Q_o/A - (L_E - L_{1E} \cdot \eta_E)}{L_{1v}/\eta_v - L_E \cdot \eta_E} \text{ for the refrigerator}$$

$$\epsilon_{3H} = \frac{Q_0/A-(L_E-L_{1E}\,.\,\eta_E)}{L_{1V}/\eta_V-L_E\,.\,\eta_{E0}} + 1$$ for the heating machine,

from which it follows after re-arrangement

$$\frac{\epsilon_{3K}}{\epsilon_{2K}} = \frac{L_{1V}-L_{1E}}{L_{1V}/\eta_V-L_{1E}\,.\,\eta_E}\left\{1+\frac{1-\eta_E}{\epsilon_{2K}}\,.\,\frac{L_{1E}}{L_{1V}-L_{1E}}\right\}$$

for the refrigerator

$$\frac{\epsilon_{3H}}{\epsilon_{2H}} = \frac{L_{1V}-L_{1E}}{L_{1V}/\eta_V-L_{1E}\,.\,\eta_E}\left\{1+\frac{1/\eta_V-1}{\epsilon_{2H}}\,.\,\frac{L_{1V}}{L_{1V}-L_{1E}}\right\}$$

for the heating machine.

Now here :

$$(L_{1V}-L_{1E})/(L_{1V}/\eta_V-L_{1E}\,.\,\eta_E) = \eta_M$$

is directly the efficiency of the machinery installation, and after further re-arrangement we obtain

$$\epsilon_{3K} = \eta_M\left\{\epsilon_{2K}+(1-\eta_E)\,\frac{L_{1E}}{L_{1V}-L_{1E}}\right\}$$ for the refrigerator

$$\epsilon_{3H} = \eta_M\left\{\epsilon_{2H}+\left(\frac{1}{\eta_V}-1\right)\frac{L_{1V}}{L_{1V}-L_{1E}}\right\}$$ for the heating machine.

If we take as an example the simplified process of a steam heat pump described in the previous chapter, then *for this process* $\eta_M = \eta_V$ and $L_{1E} = 0$, therefore

$\epsilon_{3K} = \eta_V\,.\,\epsilon_{2K}$ for the refrigerator
$\epsilon_{3H} = \eta_V\,.\,\epsilon_{2H} + (1-\eta_V)$ for the heating machine.

In the case of this simplified process it is therefore possible to introduce the adiabatic efficiency of the compressor as a simple factor in the calculation of the refrigerator. Even for the simplest of heating machines this is, however, no longer admissible.

INFLUENCE OF PRELIMINARY DEFORMATION ON THE DECOMPOSITION OF AUSTENITES ON COOLING.*

(From *La Technique Moderne*, Vol. 35, Nos. 17/18, September, 1943, p. 141.)

MESSRS. JOLIVET AND PROTEVIN have previously shown that the decomposition of austenites at a constant temperature in the upper transformation range takes place as a process of germination and growth. The graphs of quantity-changed against time show an initial period of inhibition, followed by a period of acceleration and a final slower period.

This progressive process is affected when the austenites are distorted. A load applied during the transformation period only alters this process if it is accompanied by an appreciable distortion, but a preliminary distortion of the austenites modifies the process in the absence of any load being applied during the transformation period.

This alteration is shown in a reduction, or even in the elimination, of the period of inhibition, and by a speeding-up of the period of acceleration. These changes are due to a considerable increase in the number of transformation centres (N), although the speed of growth (G) of these centres is not noticeably changed.

The curves (Figs. 1 and 2) show the progression of decomposition as measured by the change in hardness, on cylindrical test pieces of a steel containing 0.65 per cent C, 1 per cent Cr, 0.60 per cent Mo, heated to 900 deg. C. and then held at a constant temperature in the upper Ar_1 zone (720 deg. C. for curves 1 and 2, Fig. 1; 620 deg. C. for curves 3 and 4, Fig. 2). Curves 1 and 3 show the results without any preliminary deformation. Curves 2 and 4 show the results after a compression of 30 per cent, applied before being held at the constant temperature. Figs. 3 and 4, taken at points *A* and *B* on Curves 1 and 2, show, by microphotographs, the increase in the number of centres of transformation.

The influence of deformation is more marked at higher temperatures (Curves 1 and 2) than at lower temperatures (Curves 3 and 4) in the change zone, and this fact must be attributed to the relative variations of (N) and (G with those of temperature.

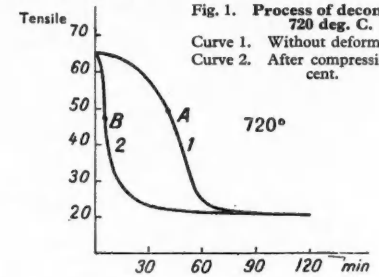


Fig. 1. Process of decomposition at 720 deg. C.
Curve 1. Without deformation.
Curve 2. After compression of 30 per cent.

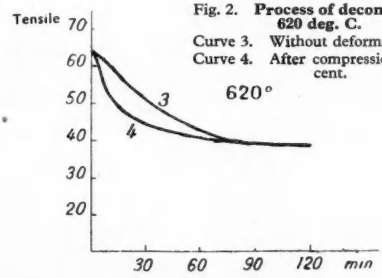
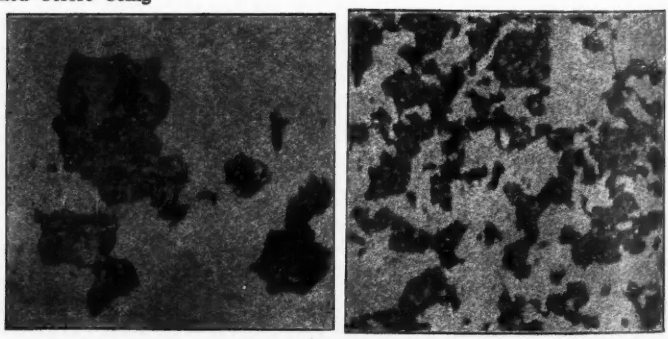


Fig. 2. Process of decomposition at 620 deg. C.
Curve 3. Without deformation.
Curve 4. After compression of 30 per cent.



Figs. 3 and 4. Micrographs (X265) at points A and B on curves 1 and 2.

*Note of Messrs. Henri Jolivet and Jean de Lacombe, presented to the Academie des Sciences by Leon Guillet. Vol. 214, No. 22

ROLLING MILLS FOR LIGHT ALLOYS

(From *Mécanique*, Paris, No. 315, July, 1943, pp. 217-220).

It has been shown that the rolls and housings of rolling mills should be of larger sizes than at present so that (a) Rolling can be done in fewer passes.
(b) A more regular thickness of sheet or strip can be assured. Regularity cannot be obtained if the rolls give too much and if the housings, adjusting screws, etc., are too flexible.

To obtain cold-rolling of most alloys, pressures and power of rolling must be high to get maximum reduction per pass. Compression of the metal increases greatly at the same time as cold deformation (more than with iron) and this means a greater number of passes, especially if the mill fittings are too small. This is especially so when the rolls are of large diameter, so that there can be said to be a maximum economic diameter of the rolls, more especially in 2-high mills. This has been remedied by using 3-high, 4-high and multiple roll mills.

Rolling conditions fix the sizes of rolls, housings, bearings, screws, etc., and the general formula used is as follows :—

$P = \sigma.b.\sqrt{R(h_1 - h_2)}$, where
P = total rolling pressure,
σ = pressure per unit area in kgs./mm.^{2},
b = width of strip or sheet,
R = radius of roll,
h_1 = thickness before rolling,
h_2 = thickness after pass.
$\sqrt{R(h_1 - h_2)}$ = horizontal projection of surface of contact between material and roll.

Values have been given for iron, steel, copper, bronze and pure aluminium (*S.K.F. Review, 1935*) but not for various light metal alloys. Some makers use Al. × 2.5, others use the Fe. figure. Emicke and Lucas have given figures for various metals and light alloys in *Metalkunde* (*1942, Nos. 2 & 3*). The figures can be used in calculating rolling pressures and driving-power needed.

In order to fix sizes of housings and gears, it is essential to know the maximum load. This can be obtained from the above formula if the coefficient *kf* is known, which can be taken as :—

kf = 25 for the largest pass, hot rolling to final thickness of 6 mm. appr.
kf = 100 for the largest pass cold rolling to a final thickness of 1 mm. appr.

Taking the case of rolls 700 mm. dia. and 400 mm. dia., and assuming $\sigma = 25$, $b = 1000$ mm., $h_1 - h_2 = 2$ mm., the total rolling pressure is

For 700 mm. roll, $P_1 = 662.5$ tons.
For 400 mm. roll, $P_2 = 500$ tons.

Thus, P_1 is 30 per cent. appr. greater than P_2 and the force required is 30 per cent. appr. greater in the first case, or, in other terms, the rolling effect is 30 per cent. weaker in the first case. These figures show why more passes are necessary when working with 2-high than with 3-high or 4-high mills.

Fig. 1 shows a range of useful sizes for 2-high mills, Fig. 2 for 3-high and 4-high mills.

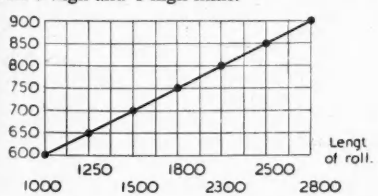


Fig. 1. Diameters of rolls for 2-high mills.

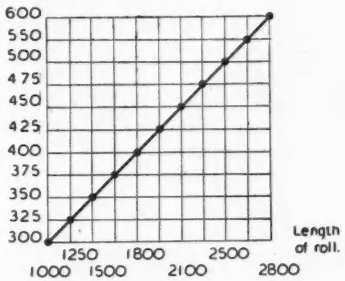


Fig. 2. Diameters of working rolls for 3- and 4-high mills.

HOUSINGS.

The thickness of the uprights is determined by the length of the necks of the rolls or by the length of their bearings so that the width must be fixed according to the load. If too small, the diagram of forces is as shown of Fig. 3. Load *P* makes the head cover bend and this applies a bending moment to the two uprights, which makes them bow towards the inside. This leads to various troubles, high number of passes, uneven thickness of rolled products. Uprights should be as wide as possible, limited by the maximum length of bearings. Housings should be wide enough to prevent any deflection, i.e., vertical deflection in the column, and the head covers must also be large enough to resist deflection. Fig. 4 shows this arrangement where the stresses are all tension, not compression-and-tension as in Fig. 3. If uprights are high elongation is greater and this leads to increase in section and so to the very great weight required in 4-high mills, 100 tons and over.

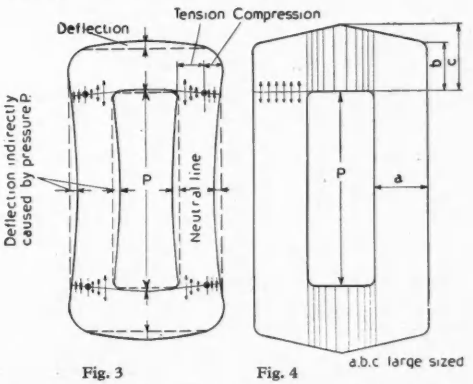


Fig. 3 **Fig. 4**

Figs. 5 and 6 show 2 different types of 3-high mills for doing the same rolling. The mill (5) has a minimum section about 17 per cent. less than mill (6) and the head covers are different. In practice, with the lighter mill (5) 2 passes more are required than with mill (6), 12 passes as against 10, or 20 per cent. more. The weights are 20 tons and 22 tons, so that a slight increase in weight gives a definite advantage to mill (6).

Taking sizes on Fig. 5, with a total rolling pressure of 1200 tons, an upright of this housing has to support a tensile stress of $\frac{1,200,000}{4}$ or, 3000 kgs., plus weight of

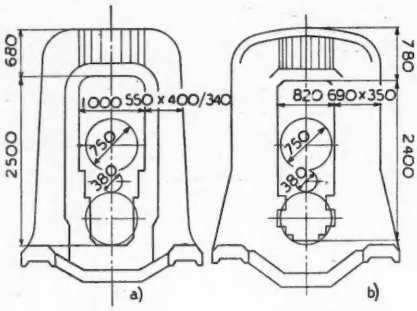


Fig. 5 Fig. 6

rolls. The elastic elongation of the upright is then :—

$$\lambda = \frac{P.I}{E.F} = \frac{300{,}000 \times 250}{2{,}100{,}000 \times 2000} = 0.018 \text{ cm.} = 0.18 \text{ mm.}$$

to which has to be added the deflection of the upper and lower covers.

Taking the section of the cover as 360×830 mm., the deflection is

$$f = \frac{P.l^3}{E.I.192} = \frac{600{,}000 \times 100^3}{2{,}150{,}000 \times 1{,}715{,}361 \times 192} =$$

$$= 0.009 \text{ cm.} = 0.09 \text{ mm., each.}$$

The total elastic elongation of the housing is therefore 0.36 mm., which is as much as the normal thickness of a rolled sheet.

Taking a 4-high mill with rolling pressure of 1500 tons, uprights $45 \times 90 = 4050$ cm^2; covers 60×125; pass 150 cm.; free length of housing column 480 cm.; elongation of the upright is then 0.16 mm. This 4-high mill is much better adapted for its load than the 3-high mill above. To improve matters more, it is necessary to go further in this direction.

The moment of rotation is $M_d = P.(a.06).2$, where P equals the rolling pressure and a the length of the pressed surface in the direction of rolling. Power is calculated from the formula

$$M_d = 71620 \frac{N}{n}, \text{ which gives}$$

$$N = \frac{M_d.n}{71620}, \text{ increasing this value by 30 per cent. to}$$

cover losses in bearings, etc.

To illustrate the case of a specific light metal rolling installation, the hot-rolling mill for slabs is a universal reversible 2-high mill with rolls 1050×2800 mm., with housings $250 \times 1000 \times 1600$ with a weight of about 1000 kgs. Behind the 2-high mill at about 50 metres distance, is a 4-high mill with rolls 1300/500/500/1300 for the final pass. Owing to the small diameter of the support rolls a considerable reduction in thickness is obtained and an even thickness and smooth finish on the rolled strip of 6.5 to 7 mm. thick.

By using three or four 4-high mills instead of one, the thickness can be brought down to 3.5 mm. and the amount of cold working is thus reduced. The cost of 4-high mills is, however, a serious item. Production can be increased by replacing the 2-high by a 4-high mill with rolls 1500/750/750/1500 mm.

The housings are fitted with special fitments, e.g., each roll has an arrangement for giving the temperature of the roll in the middle and at the two sides, this being important for gauge and curvature. Gauge can be regulated by flame heating, or by air or liquid cooling. Each roll is also fitted with an instrument showing the curvature so that the rollers can adjust this as required.

When the roll length increases the diameter of the working rolls also increases, in 3, 4 or 6-high mills to give greater resistance as the bending moment gets bigger. This reduces the advantage of the multiple mills over the duos. For hot-rolling mills, roll diameter depends upon the thickness of the pass and on the angle of attack, so that for 2800 mm. wide aluminium slabs, the working rolls may be 730 mm. diameter. Although large, these rolls are still much smaller than in a 2-high mill of the same width.

The following table gives diameters of rolls for a series of 4-high mills :—

Width :	1050 mm.	diameter :	1000/450/450/1000
,,	1250 mm.	,,	1100/450/450/1100
,,	1700 mm.	,,	1200/600/600/1200
,,	2000 mm.	,,	1350/650/650/1350
,,	2500 mm.	,,	1350/700/700/1350

As the thickness of the slab decreases at each pass and its factor of adhesion is thereby changed, the working rolls ought, theoretically, to decrease in size at each pass to get the full benefit of rolling with small diameter rolls. This is only possible when housings are placed one behind the other.

COLD ROLLING.

The question of roll diameter is more important in cold rolling, where the most serious rolling is done. Examples of rolls are :—

3-high	Width : 1200 mm.	dia. :	550/350/550 mm.
	,, 1800 mm.	,,	700/380/700
	,, 2400 mm.	,,	750/420/750
4-high	,, 1450 mm.	,,	1240/450/450/1240
	,, 2500 mm.	,,	1350/480/480/1350
	,, 2800 mm.	,,	1400/500/500/1400

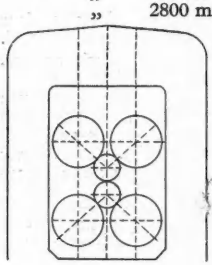


Fig. 7. **6-roll mill.**

A roll 480 mm. diameter; a reduction in thickness of 1.5 to 1.2 mm. in one pass: kf of 80 kg./mm.^{2} and a width of rolling of 1050 mm., give a calculated rolling pressure of 714,000 kgs. With a roll 240 mm. dia., rolling pressure, ceteris paribus, is only 512,000 kgs. Reduction in thickness per pass corresponding to the higher rolling pressure (714,000 kgs.) will increase to 0.58 mm. as

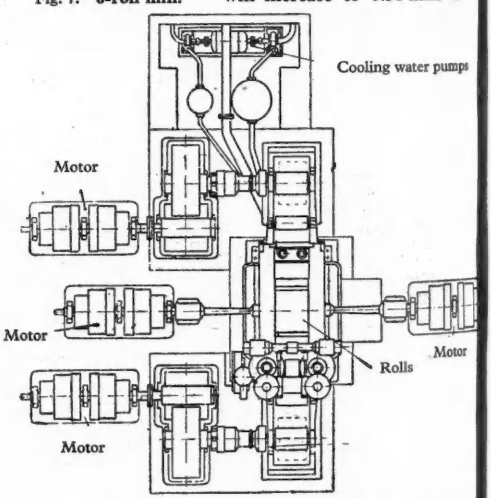


Fig. 8. **20-roll mill for strip up to 1000 mm. wide.** Working rolls 90 and 160 mm. dia. Support rolls 850 mm. dia.

against 0.3 mm., that is nearly twice as large with the smaller rolls as with the larger rolls, the usual formula being used,

$$P = kf.b. \sqrt{R(h_1 - h_2)}$$

This phenomenon is explained by the "wedge action" exerted by differing sizes of rolls. Theoretically, the rolling effect of a 3-high mill with only 1 small rol is only half that of 4-high mill.

Efforts to use as small rolls as possible have led to the designing of various types of mills, for instance a 6-high mill for rolling medium width strip. The 2 working rolls are each supported by 2 support rolls, as shown in Fig. 7. This mill must be very accurately built to ensure perfect contact between the working roll and its two support rolls. One rolling mill maker has made it possible to use small rolls on a 4-high mill by the use of special supports. Rolling is only done in one direction and the rolled strip proceeds under the mill ready for a further pass.

Multiple roll mills for thin alloy steel strip have been designed by Dr. Rohn with 12 to 20 rolls. Fig. 8 shows a plan of a 20-roll mill for iron strip up to 1000 mm. wide. Rolling is done in 5 passes, reducing the thickness from 1.8 mm. to 0.2 mm. This mill is also suitable for rolling light alloys as the small diameter of the working rolls allows for :—

1. Large reductions in thickness per pass without intermediate annealing, hence
2. Large production, hence
3. Large saving in power
4. Reduction in costs.
5. Most satisfactory rolling of very thin strip.

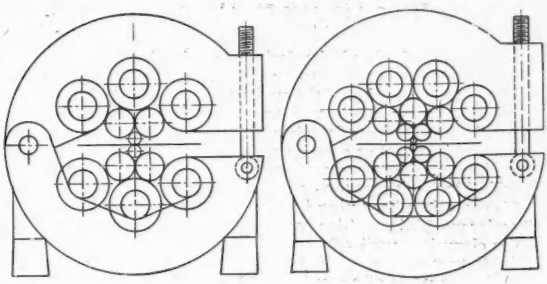


Fig. 9 Fig. 10

Fig. 9 shows the arrangement of the rolls in a 12 roll mill.

Fig. 10 shows a 20-roll mill.

These multiple roll mills will find a place as finishing rolling mills for light metals, but, for a reducing mill, the 4-high mill is more suitable, if only for its lower first cost.

NITRIDING STEELS FOR LARGE TOOLS FOR PLASTIC MOULDING.

By Dr.-Ing. W. Haufe, VDCh., Düsseldorf. (From *Kunststoffe*, Vol. 33, No. 6, June, 1943, pp. 151-158).

Developments in the plastic moulding industry during recent years show an increased demand for large moulding tools. Whilst the hardening of small or medium size tools requires no serious precautions, considerable difficulties are experienced with larger tools. Distortion and very often cracking occurs on tools above 350 sq. mm. sectional area. It is general practice, therefore, to use large tools without being hardened at all.

The strength of the steels used for large tools, self-hardened or normalised, varies from 60-120 kg./sq. mm. For high resistance against wear, steel of 90-100 kg./sq. mm. strength is very often used, the top limit for machining (i.e., special milling) being 120 kg./sq. mm. The influence of alloy-elements upon the mechanical properties are shown in Fig. 1. The advantage in using these materials lies in the fact that high accuracy can be maintained without the tool being hardened but, after a certain run, scoring occurs on the surfaces where the excess material escapes when moulding. When high specific pressures are applied galling occurs on the surface of the tool resulting in depressions on the surface of the finished moulding. The appearance of the finished product is thus affected and the use of unhardened tools should be avoided whenever possible.

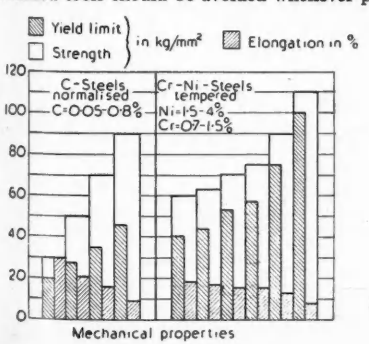


Fig. 1

Distortion and cracking in hardening is caused by stresses occurring during heat treatment. When cooling down the contraction of the outer portions of the tool is hindered by the relatively hot core and cracking occurs when the strength is not sufficient (at the known temperature). With further cooling, the contraction of the core results in tensile stresses in the core, and compression on the outer portion. Additional stresses are present where there are uneven wall-thicknesses as the result of non-uniform volume changes.

All stresses depend on the temperature gradient and therefore on the size and shape of the tool, and the cooling rate of the quenching media.

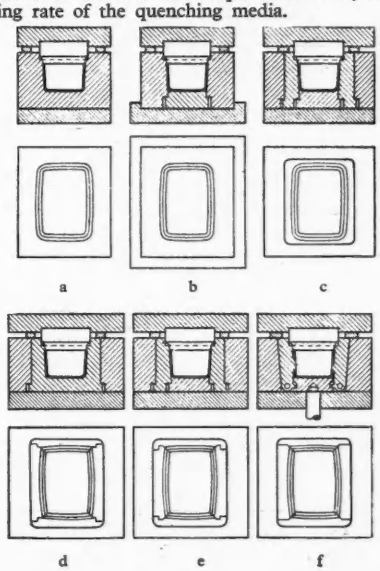


Fig. 2

INFLUENCE OF SHAPE ON THE HARDENING-STRESSES.

Large tools should be designed as simply as possible with no sharp corners and with an even material distribution. In spite of the various shapes of mouldings there are some basic tool types and it is possible to consider the safety of the hardening process on the basis of a cabinet tool as shown in Fig. 2.

Top and bottom tool can be machined from the solid block without too deep a cavity (Fig. 2a). In Fig. 2b there is a base plug inserted in the bottom tool for easier machining. The bottom press-platen is machined for clamping, whilst 2a and b are solid block tools, the other types c-d-e are " built up " tools with a chase to shrink-fit parts of a mould cavity in place, or as in " f " to enclose a hinged assembly which is ejected with the molding.

The heating method during moulding has an influence on the hardening, too. Tools with bores for heating elements or channels for steam are the more dangerous and it is very difficult to avoid cracks in such tools. Whilst " built up " tools are easier for hardening, it is obvious that design considerations and tool construction are not sufficient for safe and distortion-free hardening.

The hardening process itself is determined by the steel-quality which should be in accordance with the general requirements of plastic moulding tools:

Highest resistance against wear
High polish
Sufficient strength
Good tenacity and shock resistance.

Alloy-steels complying with these requirements and with standard Government specifications are shown in Table I.

There is very little information regarding the stresses which occur during the hardening and tempering of large forgings. Quenching in oil results in approx. $\frac{1}{2}$ to $\frac{1}{4}$ of the stresses set up when quenching in water. The stress when cooling in air is about 1/10 and when cooling down in the furnace is 1/20-1/50 of the stress compared with water quenching.

The assumption that large tools air-hardened and air-cooled are distortion-free is not correct. The treatment of a bottom tool machined from a solid Cr-Ni-Mo steel block, hardened at 880 deg. C., with ventilated-air and tempered at 200 deg. C., resulted in a strength of 125 kg./sq. mm., which is an insignificant improvement in comparison with self-hardening steel. There is considerable scaling, and much polishing is required.

Carburising is recommended as a safe method for hardening all medium size tools, but with large tools, only simple shaped designs are distortion and crack free. There is little information regarding hardening of large tools in baths.

NITRIDING.

Nitriding, like carburising, is a surface hardening process. Table 2 shows standard nitriding steels complying with Government specifications.

The nitriding process consists of heating the steel in an atmosphere of ammonia gas at a temperature of 500 ± 10 deg. C. At these temperatures an absorption of nascent nitrogen occurs, which produces on the surface a case of extreme hardness. Elements which form nitrides in steel are aluminium, chromium, vanadium and titanium. The depth of the nitrided case depends on the length of time of nitriding and the maximum depth is approximately 0.8 mm. Fig. 3 shows the functional graph between time and depth. The nitriding time for standard moulding tools is 70-100 hours, but components with thin sections, like toothed gears, are only heated 10 hours to avoid through-nitriding of the thin sections.

The properties of the nitrided case are:

High surface hardness
High resistance against wear
Certain resistance against corrosion
High fatigue and corrosion fatigue resistance
High resistance to notch effect.

The surface hardness from nitriding is superior to all other known hardening processes and ranges between 900-1100 Vickers. Fig. 4 allows a comparison with standard hardening processes and steels.

The exceptional hardness 30-50 per cent above the surface hardness of case-hardening results in

TABLE I.

Item	Hardening Process	Max. Permissible Content on Alloys				Average Values of Standard Steels						
		Cr	Mo	Ni	V	C	Si	Mn	Cr	Mo	Ni	V
1	Case hardening	1,5	0	0	0,20	0,20	0,30	1,40	1,40	—	—	—
2	Oil or air hardening (a)	1,3	0,3	0	0,25	0,50	0,70	0,90	1,25	0,25	—	0,20
	(b)	2,3	0,5	0	0	0,45	0,25	1,75	2,00	0,25	—	—
	(c)	0,8	0,5	1,8	0	0,55	0,40	0,50	0,70	0,45	1,70	—
3	tempered or natural	1,3	0,3	0	0,25	0,60	0,30	0,70	—	—	—	—
		2,3	0,5	0	0	0,50	0,70	0,90	1,25	0,25	—	0,20
		0,8	0,5	1,8	0	0,45	0,25	1,75	2,00	0,25	—	—
4	Nitriding Process	Mn 0,8	Cr 2,5	V 0,25	Mo 0	0,30	0,40	0,60	2,40	—	—	0,25

TABLE II.

Nitriding Steel Type	Composition approx. Cr	Al	V	Strength Properties: Yield limit kg/mm²	Strength kg/mm²	Elongation (l=5d) %	Reduction %	Application	Remarks
1 Cr-Al	1,3	1,0	—	45	65-80	18	55	Up to 60 mm. dia. or square	—
2 Cr-Al	1,2	0,9	—	45	65-80	14	45	Up to 60 mm. dia. or square	for injection tools where rather good machinability than strength is required, e.g., for comb-tools
3 Cr-Al	1,3	1,1	—	60	80-100	14	45	Up to 100 mm. dia. or square	—
4 Cr-V	2,4	—	.20	70	90-100	14	50	for all sizes	for all tools where high strength is required
				80	100-115	12	45	for all sizes	

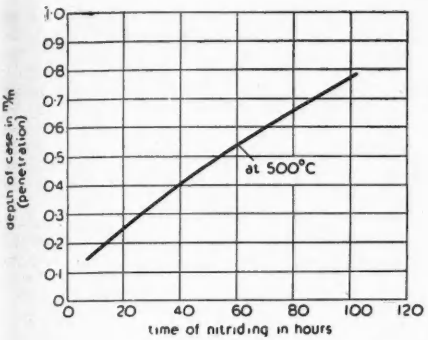


Fig. 3

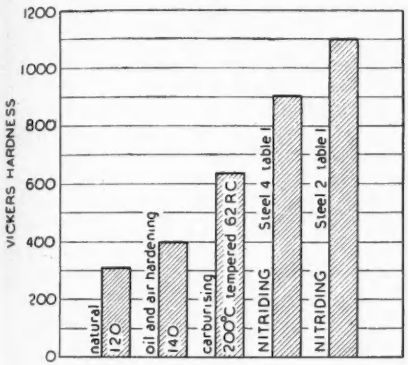


Fig. 4

extremely high resistance against wear, and no flow or scoring marks were noticed on nitrided tools after comparison tests with other tools. Comparison with gauge steel and chromium plating carried out on plug gauges gave the following results in numbers of parts gauged :

Gauge steel	16500
Chromium plated	32500
Nitrided steel	53500

The high resistance against wear entails a high polish suitable for long runs, and makes the appearance of the mouldings perfect.

Experiments to ascertain whether nitriding steels would withstand the chemical action when pressing urea plastics are not conclusive, but they can be chromium plated without any difficulty.

NITRIDING STEELS FOR MOULDING TOOLS.

Tools to be nitrided are usually normalised and drawn, or quenched and drawn to the desired hardness and ductility required in the core. With regard to machining it is not recommended to go beyond a strength of 110 kg./sq. mm.

For large tools as used for pressing panels for radio-cabinets, nitriding steel of 90-110 kg./sq. mm. tensile strength is recommended. The yield point is about 90 to 70 kg./sq. mm. With cabinet mouldings of 900 sq. cm. projected area 360 tons pressure is usually required, i.e., 4 kg./sq. mm. which gives a safety factor of 18. In extreme cases, with uneven moulding powder distribution, the total tonnage may be acting upon a smaller surface within the tool-moulding area, but in no case would this result in higher specific pressures than the yield strength of the steel. A steel with 90-100 kg./sq. mm. strength must be satisfactory under normal moulding conditions and all experiences have endorsed this.

Nitriding steel used for intricate shapes should be normalised prior to finish machining.

The finish machined tools are packed in a gas tight container and hardened in a furnace (Fig. 5). After the hardening time of three to four days with large tools, the cooling down proceeds slowly in the ammonia gas atmosphere. Afterwards the tools are in the hardened state and are ready for use as moulds. There is a slight increase in sizes up to an amount of .02-.03 mm. which can generally be neglected.

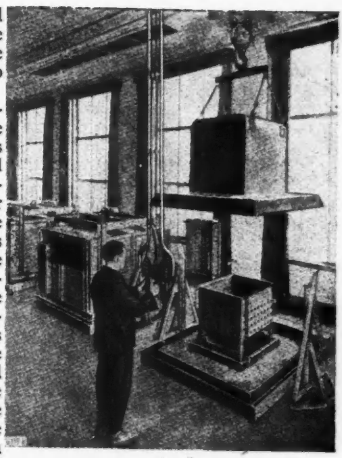
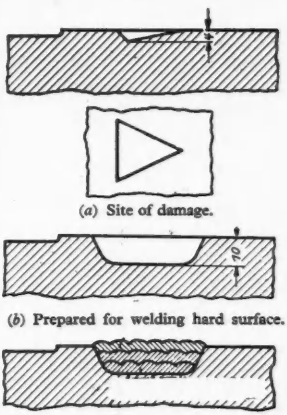


Fig. 5

In spite of the long heating and hardening time, the nitriding method is economic because finishing work, after hardening, is avoided and no tools need be scrapped due to cracks.

It is sometimes assumed that surface damage on nitrided tools cannot be repaired. Fig. 6 shows how the bottom of a nitrided tool was repaired by welding. The damaged part was ground to 10 mm. depth and the tool heated up to 280-300 deg. C., and a hard material, approximately 500 Brinell, was welded in.



(*a*) Site of damage.

(*b*) Prepared for welding hard surface.

(*c*) Welded. Fig. 6

FLAME CUTTING OF STEEL.

By P. GRASSMANN, *VDI*, and R. BECHTLE, *VDI*.
(From *Zeitschrift des VDI*, Vol. 87, Nos. 37/38, September, 1943, pp. 603-604.)

IN order to be able to judge whether the speed of flame cutting can be further increased, it is necessary to investigate the various phenomena which occur successively in the cutting process proper, and which of these takes place at the lowest speed, hereby determining the speed of the cutting process as a whole.

Flame cutting is characterised in that the oxygen jet of the torch oxidises the iron and then blows the molten oxide out of the cut thus made. At the high temperatures concerned, the oxidation process proper takes place very rapidly ; however, as soon as a state of oxidation has been established, a slag layer forms on the surface of the material which prevents any further immediate contact of the oxygen with the metal. This reaction can therefore only continue to the extent to which solution or diffusion processes make it possible for the oxygen to reach the iron through the slag layer, or vice versa, for the iron to rise through the slag film to the surface. Any slag removed by the action of the

oxygen jet is at once replaced, so that the surface of the iron remains permanently covered with slag.

If cutting is carried out with impure oxygen, e.g., containing nitrogen, a further difficulty comes into play. As only the oxygen enters into chemical combination a progressive nitrogen enrichment is taking place in the slag surface, and this nitrogen-rich surface layer is mainly responsible for the smaller cutting speed obtainable with impure oxygen. Since, however, in most cases oxygen of 99.3—99.9 per cent. purity is used, this reaction-retarding effect due to the presence of nitrogen may be neglected.

The phenomena taking place in the cut itself may be described as follows: Since for each centimeter of depth of cut identical amounts of steel must be converted into slag, the amount of slag descending in the cut must increase towards the bottom of the cut, and the thickness of the slag layer must therefore also increase towards the bottom, as indicated in Fig. 1. This leads to a slowing down in the speed of reaction between iron and oxygen. Since the diffusion process takes place in a layer zone of laminar flow, the speed of reaction can be assumed to vary inversely with the layer thickness. Therefore, reaction speed will be very high in the upper part of the cut; but here also the oxidation cannot proceed faster than in accordance with the cutting speed. Thus in the upper part of the cut the cutting line through the sheet will be a straight extension of the oxygen jet. Further below in the cut the cutting line shows the characteristic point *A* of inversion (Fig. 1), in which the greatest speed of reaction possible with the prevailing thickness of slag equals the mechanically imposed speed of travel of the tip of the torch through the material, that is, the cutting speed. Below this point, the reaction speed is less than the rate of feed as exemplified by the progressively slanting course of the cutting tracks. However, the greater is this slant, the larger is the area of contact between oxygen jet and material. The resulting slant, expressed by the angle of incidence α will therefore be given by $v = v'/\cos\alpha$, where v is the rate of feed and v' is the speed of reaction. Since v' decreases with the thickness of the layer of slag, the angle α will, therefore, increase with the depth of cut.

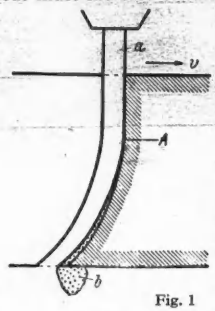


Fig. 1

In order to obtain a clean cut, however, angle α should not differ much from a value of 90 degrees. Thus, there exists a limit condition for the relationship between thickness of sheet "shearing" power of the jet and the rate of feed. To establish a functionality of this kind, certain assumptions would have to be made regarding the viscosity of the slag and the shearing power of the jet, etc. At the present time, nothing is known with regard to these factors; and a relationship in full conformity with actual conditions cannot yet be established. But even so, these considerations may well be able to serve as the starting point for the development of an analytical treatment of the flame cutting process.

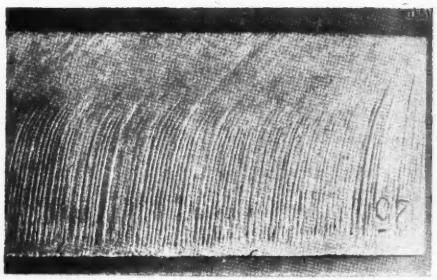


Fig. 2. **Steel of 55 mm. thickness cut with considerably too high cutting speed.**

A RUBBER-SAVING EXPERIMENTAL TYRE

By Dr.-Ing. H. P. Zoeppritz, VDI. (From *ATZ Automobiltechnische Zeitschrift*, Vol. 46, No. 9-10, 25th May, 1943, pp. 225-231.)

If the tyre pressure of giant tyres is increased the deformation under load is reduced and especially the compression of the profile surface, and deflection of the tyre walls. Compared with this state of affairs, some reduction of springing characteristics and increase of bending stresses in the tyre shoulder, caused by partial replacement of the tyre side walls by rigid walls, should be permissible. Following these lines of thought, the Stuttgart Automobile Research Institute has designed a material-saving giant tyre of size 12.00–22 (Fig. 1), with shortened side walls. The volume of air is the same as that of the original design, but half of the original walls is replaced by the extended rim. Whether a still more radical shortening of the tyre sides is possible, remains open to further research.

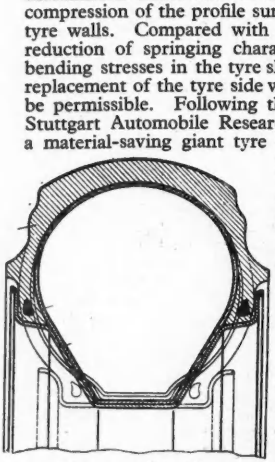


Fig. 1. **Cross-section of a tyre with reduced walls, size 12.00-22, in comparison with a normal giant tyre of same size.**

An experimental car tyre, size 5.50-16, has been constructed on the same principle with the aid of the Central Rubber Laboratory of I. G. Farbenindustrie, at Leverkusen. Its characteristics have been tested. Its weight is 16 per cent. less than that of a standard tyre, and its rolling resistance 10-20 per cent. smaller dependent on air pressure and load. The lateral guiding forces are much higher as the walls are supported by a broader rim, while the resilience for obvious reasons, is smaller. The tyre has so far undergone some 1,000 km. in road tests, but neither these tests nor the development work is completed.

Fig. 2. **Tyre, size 5.50-16, with reduced wall (1), and standard tyre (2).**

LIGHT-WEIGHT CONSTRUCTION IN RAILWAY AND ROAD VEHICLES

By DIPL. ING. TH. CLEFF, VDI. (From *Zeitschrift des VDI*, Vol. 87, No. 25-26, June 26th, 1943, pp. 377-384.

GENERAL PRINCIPLES IN THE DESIGN OF LIGHT-WEIGHT VEHICLES.

THE two governing principles are:

1. Stressed-skin construction in the design of closed vehicles.
2. Application of fatigue data and regard to the effect of shape on strength.*

APPLICATION TO DESIGN OF KNOWLEDGE ON FATIGUE STRENGTH.

THE usual diagrams of fatigue strength under reversed load show grater allowable compression stress than tensile stress. Thus, the usual spring material has a 30 per cent. greater permissible compression than tensile stress. The efficient design of a leaf spring, therefore, is dependent on this fact, and is obtained by shifting the neutral axis toward the tension side, Fig. 1.

Fig. 1. **Shift of the neutral axis of a leaf spring toward the tension side : e_z height of tension side, e_D height of compression side.**

The same experience is utilised in the design of buffer springs for railway purposes (Fig. 2) where the inner compression spring is of lesser dimensions than the outer tensile spring. It is frequently overlooked that the fatigue strength is dependent on the cross-sectional dimensions, i.e., that the smaller cross-section is relatively the stronger one† Thus, replacing a helical compression spring (Fig. 3) with a free length of 470 mm. by four smaller springs (Figs. 4 and 5), 47 per cent. of weight can be saved. The four smaller springs require no more space than the larger single spring.

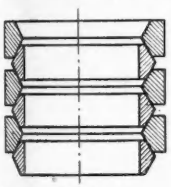
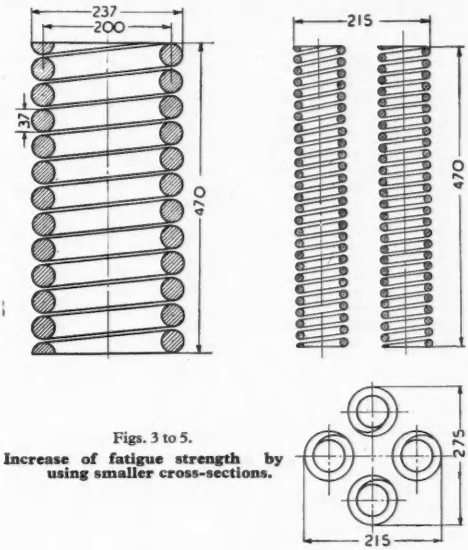


Fig. 2. **Weaker inner ring (compression) of a ring spring.**

The foremost task is to avoid stress-raising notches, i.e., to adopt designs that offer the least possible notch-effects. Investigations carried out with steel U-beams have shown that holes and rivets in the tension flange reduce the fatigue strength considerably. Weld seams in the tension flange reduce the fatigue strength even more pronouncedly. These effects are absent on the compression side. Fully welded frames for vehicles cannot therefore be recommended, unless each local cross-section is designed so that constant fatigue strength is guaranteed. Such a fabricated structure is shown in Fig. 6. It is a bogie frame for a German railway car; the frame is very light, but, of course, the production time is considerable. Welding itself leads to very stiff structures, and for high fatigue strength the joints must be carefully designed. Fig. 7 shows a welded joint of two U-sections in accordance with latest experience with respect of fatigue. (Note the contour of the gusset plates).

Notch effects cannot be excluded entirely by design, as they can arise through imperfections in workmanship. Again, the superposition of internal stresses can lead to increase in fatigue strength. This has been utilised in the production of leaf springs. The spring leaf is to be prestressed in flexure to an extent by far exceeding the yield point. In the centre of the cross-section the material remains elastic, the outer fibres, however, being permanently deformed cannot regain their original length, so that on the tension side of the spring there is a residual tensile stress in the centre, and a compression stress in the outer fibres (see Fig. 8). The shape of the leaf is shown as inset to Fig. 8.



Figs. 3 to 5.
Increase of fatigue strength by using smaller cross-sections.

INCREASING THE RESISTANCE TO DYNAMIC LOADING.

Vehicle parts must be so designed as to withstand shocks without permanent deformation; that means a high ability to absorb strain energy. The greater the

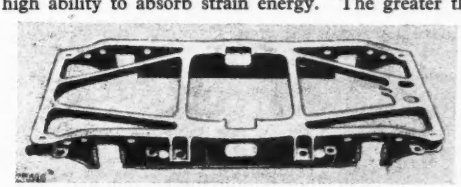


Fig. 6. **Fabricated bogie for a corridor car.**

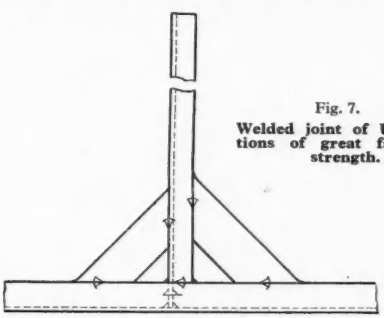


Fig. 7.
Welded joint of U-sections of great fatigue strength.

* See A. Erker: *Werkstoffausnutzung durch festigkeitsgerechtes Konstruieren, Z.V.D.I.*, Vol. 86 (1942), pp. 385-395.

† W. Buchanan: *Einfluss der Querschnittsgrösse auf die Dauerfestigkeit, Z.V.D.I.*, Vol. 87 (1943), pp. 325-327.

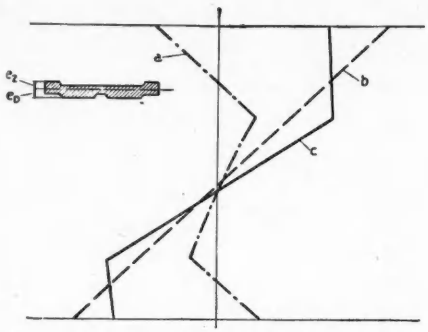
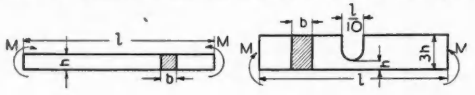


Fig. 8. Stress systems in a prestressed spring leaf.
(*a*) Internal stresses due to plastic deformation ; (*b*) Estimated stress due to bending moment ; (*c*) Resultant stress from *a* and *b*.

strain of a material the smaller the forces and stresses transmitted. Under dynamic load the requirement is to absorb shocks within the elastic range of the materials ; moreover in accidents to provide the greatest possible plastic deformation. Thus, the requirements can be formulated :

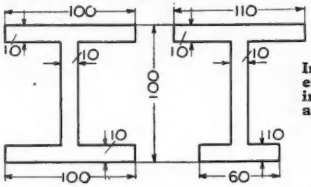
1. Materials of high strength and yield point.
2. Application of fatigue data to design in the light of knowledge on the effect of shape on strength. In particular this tends towards design of parts of constant strength.

The peculiarities of design for dynamic loading can be illustrated by a few examples. Two sections (Figs. 9 and 10) exhibit under static load the same strength.



Figs. 9 and 10. Example of efficiency of material under impact load.

Under dynamic load, however, the beam of height 3 h has only 2/15 of the strength of the constant section beam. Similarly, with I-beams the flexural strength can be increased by asymmetrical design. Thus, the beam in Fig. 12 has the same modulus of section as the beam in Fig. 11, with a 10.7 per cent. smaller weight and 21.5 per cent. greater dynamic strength. As the modulus of section is the same, the two beams are of equal strength under static load ; the beam in Fig. 12 has, however, a smaller moment of inertia than that of Fig. 11, consequently the asymmetrical beam can absorb a 21.5 per cent. greater strain energy.



Figs. 11 and 12. Increase of strain-energy of an I-beam in flexure through asymmetrical cross-section.

CHASSIS DESIGN.

The chassis or underframe, as the heaviest item of a vehicle, is quantitatively the most promising from the weight-saving point of view.

FREIGHT CARS.

The underframe of a two-axle freight car has to withstand the following loads :

1. Bending moments and shear forces due to the loading and to its distribution on the springs.
2. Torsion due to unevenness of the track.
3. Buffing forces, which can be symmetrical or asymmetrical.

Fig. 13 shows the underframe of the "Ommu" wagons of the German State Railways, made of rolled sections. Instead of heavy longitudinals there is a

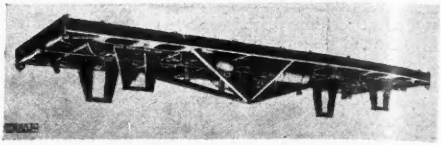


Fig. 13. Under frame of the "Ommu" freight car.

lattice framework which takes also the shear loads and torsion. This frame is so light that with two 20-ton axles a pay load of nearly 30 tons can be carried. With covered vehicles the aim is to avoid the use of the underframe entirely, and make the body carry a proportion of the load by utilising a stressed skin construction. Similarly, in tank cars, the actual tanks have been made load-carrying, thus eliminating the underframe, which hitherto was the load-carrying agent, with the tank simply resting on it. Such tanks can be built with wall-thickness of 6 mm. the bottoms with 8 mm. thickness. Essential to the application of this type of design was the solution of the problem of suitably spreading the applied loads. For example, the Deutz tank (Fig. 14) shows the headstocks which diffuse the heavy traction and impact loads. The tank shown is put directly on the bogies.

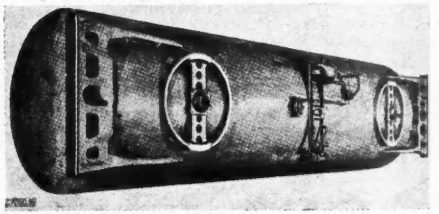


Fig. 14. 4-axle Deutz-type tank car of 63 cu.m. capacity.

Another similar type of vessel is the Uerdingen, Fig. 15. In the vicinity of the buffer axes thin-walled sections welded to the bottom of the tank serve to diffuse the impact loads.

In spite of the great weight saving effected—the empty weight per cu.m. of tank capacity decreased by 49 per cent. with four-axle cars—under buffing loads up to 200 tons, the stresses nowhere exceeded the allowable limits.

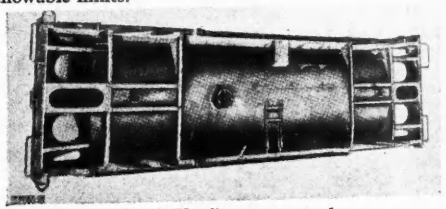


Fig. 15. 2-axle Uerdingen-type tank car.

PASSENGER CARS.

The same principles as with freight cars had to be embodied in passenger car design, i.e., making the walls and roof load carrying in shear and compression. The compressive strength of the roof is of significance in accidents, because it keeps the structure in alignment and thus contributes to localizing the damage to the end parts of the vehicle, which in case of destruction are not so likely to injure passengers.

As compared with previous methods of design this type of structure enabled a weight reduction from 40

tons to 27.8 tons for a third class corridor vehicle without any change in comfort or equipment. The whole structure, including the outside panels, was made of steel. The stiffeners can be of the stringer section type or of pressed construction. Prior to buckling of the sides, the effectiveness of the stiffeners depends on their moment of inertia and on the Young's modulus of the material. Subsequent of buckling, and during plastic deformation (accidents), the shape of the stiffeners and the strain energy of deformation is of importance. Table 1 shows test results on body sections of equal weight. It is recommended to chose the stiffener moment of inertia so that the stiffener buckles before the side plate buckles. As a typical light-weight car the Uerdingen type corridor car is shown in Fig. 16. All rules of light-weight design

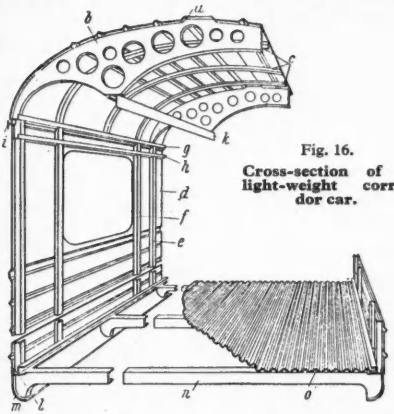
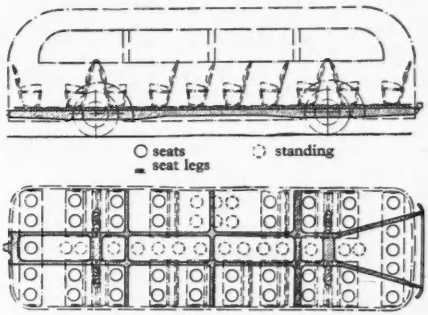


Fig. 16. **Cross-section of a light-weight corridor car.**

have been conscientiously observed, especially with respect to spreading of the buffing loads which have the test value of 200 tons at each end of the car. The floor is made of flat-sided corrugations, as it has a higher buckling stress than the usual corrugated sheet. As the energy transmitted by one car to another in case of a collision is proportional to the mass, it follows that a light-weight train has advantages in this direction.

TABLE 1

SHAPE OF SHELL AND STRINGERS	SKIN THICKNESS mm	BUCKLING STRESS kg/mm	FAILING STRESS kg/mm	TYPE OF FAILURE
	0·4	2·1	9·5	COLUMN BUCKLING
	0·4	1·7	9·9	COLUMN BUCKLING SUBSEQUENT TO TORSIONAL BUCKLING
	0·6	2·9	5·4	COLUMN BUCKLING DUE TO BUCKLING OF SKIN
	0·4	1·2	4·0	LOCAL BUCKLING OF STRINGERS INITIATED BY FOLDS IN SKIN
	0·4	1·6	12·0	COLUMN BUCKLING SUBSEQUENT TO TORSIONAL BUCKLING
	0·4	1·5	11·3	
	0·4	1·0	13·4	LOCAL BUCKLING OF STRINGERS INITIATED BY FOLDS IN SKIN



Figs. 17 and 18. **Light-metal bus trailer for 58 passengers.**

THE USE OF LIGHT ALLOYS.

The use of light alloys can be recommended only in cases where a drastic weight saving over a light-weight design in steel is possible, and where the use of light metal is economically justifiable.

The unfavourable aspect of light metals lies with their reduced strength under alternating loads. This fact reduces any gain in weight over steel beams, or at least makes such gain of dubious value. However, in body structures light alloys are superior to steel. If, to avoid buckling, the stiffness of the side plate is EI (E=Young's modulus, I= moment of inertia), it is necessary when changing from steel to light alloy to keep the value Et^3 constant (t=sheet thickness). The low modulus of light alloys makes it necessary, therefore, to increase the sheet thickness. For equal strength of sheet the weight saving with aluminium alloys is about 50 per cent. with magnesium alloys about 60 per cent.

For such reasons a mixed type of construction is advantageous. As an example, a bus trailer for 58 passengers is shown in Figs. 17 and 18. The empty weight is 3,400 kg., or 58.5 kg. per passenger. This means 30 per cent. weight saving over the steel structure. Smaller size tyres and a floor height of 600 mm. are the direct benefits of the lighter design. The floor is of corrugated aluminium alloy, the sides and the roof of magnesium alloy, the longitudinal and lateral beams forming the chassis are of steel (shaded in Figs. 17 and 18). The magnesium alloy sheet of 1.5 mm. thickness is on account of its low modulus of elasticity a very good noise reducer. Whilst with such short vehicles as the one described light alloys offer advantages, long vehicles such as a railway bogie vehicle cannot employ light metal because of the danger of resonance vibrations.

A NEW METHOD FOR WELDING CABLES

By H. GÜNTHER, VDE, Berlin. (From *ETZ Elektrotechnische Zeitschrift*, Vol. 63, No. 49-50, Dec., 1942, pp. 587-591.)

AT present two methods are used to join stranded cables and wires: soldering and brazing, or, alternatively, application of various clamps or clamping devices. This article deals with a new method by means of which welding is employed as jointing medium.

In the following the term " cable " will be used in its widest sense, and will cover electrical conductors as well as cables for purely mechanical use, consisting of at least two wires.

Cable ends can be welded together either by welding the individual wires one by one, or by butt welding the cable as a whole. The first method is generally impracticable, and the second presents a number of problems which require special care, namely :—

(A) Single thin or stranded wires of the cable must not be allowed to burn nor to oxidise, which may easily occur due to the heavy welding current required for the whole cross section of a cable.

(B) Welding heat must not be allowed to spread to such an extent as to effect the rubber, or other insulation of electrical cables.

(C) The final diameter of the weld must not unreasonably exceed the diameter of the cable, which is important in electrical cables where the joint is usually covered by one or more layers of insulating material.

(D) The length of the weld must be kept at a minimum.

All these requirements are met by the method described below.

THE NEW METHOD.

The blank cable ends to be joined are inserted in a ceramic tube whose inner diameter is 10 per cent. larger than the diameter of the cable. Both ends meet at the centre of the tube. Subsequently, a suitable alternating current is applied which, as in an ordinary butt welder, causes the extreme ends of the cable to melt. Now one cable is fed continuously into the tube which is ultimately filled with the molten material. If the current is switched off at the right moment the required portion of the tube will be filled with an homogenous block of the same material as the cable, in which both cable ends will be solidly embedded. The length of the weld is merely a function of the length of the cable fed into the tube during the welding process.

The special feature of this method is that it consists of a butt welding process in an enclosed space, and that it can be regulated automatically. This makes possible the use of heavy welding currents which in turn results in short welding times. Combined they give the following results :—

(*a*) The heavy current immediately develops considerable heat at the point of contact, while the temperature drops rapidly towards the clamps holding the cable. This is due to the electrical resistance being highest at the point of contact and the heat dissipation lowest at the centre of the tube.

The drop in temperature near the clamps is in turn further accentuated during the welding process by the higher resistance, and consequently higher heat generation at the junction points which results in the bulk of the electrical energy being converted into heat in a very limited space and close to the point of contact where the specific resistance of the material, due to the very high temperature, is a multiple of the corresponding figure for the cold solid conductor.

(*b*) The parts of the wires adjoining the molten material are at glowing temperature for a comparatively short time, whilst the temperature of the clamp remains low throughout the process and the temperature of the cables beyond, is not increased.

(*c*) Oxidation in the weld proper and in the adjoining parts is negligible, if it occurs at all and the mechanical properties of the cable are not deteriorated as a whole. Chemicals to prevent oxidation, as used for soldering and brazing, or any special soldering materials are not required.

(*d*) Arrangements or devices to prevent the atmosphere from reaching the welding spot are unnecessary. The clearance between the ceramic tube and the cable can be made comparatively large, thus enabling cables of equal cross section, but of slightly larger diameter—as may occur in production—to be welded in tubes of standard normal size.

(*e*) The ceramic tube requires no high degree of accuracy and need not therefore be ground. It must, however, consist of material with high resistance to wear and to changes of temperature.

2. THE WELDING OPERATION.

(*a*) *Description of the welding set.*

The main parts of the set are: Magnetic switch, Transformer, Welding arms (with clamps), Ceramic melting tube, and Mechanical gear with operating lever.

The magnetic switch connecting the set to the mains is energised through an auxiliary contact operated mechanically. It is switched off automatically through the auxiliary contact as soon as the required length of the cable has been fed into the melting tube.

The secondary circuit of the transformer closes over the welding arms and the cable ends which make contact in the melting tube. Transformer leakage can be kept as low as is convenient and due to the short period of operation no allowance need to be made for heating. In the lay-out of the transformer, only the ratio and the inner resistance, which is determined by the requisite intensity of the welding current, need be taken into account.

The cable ends are clamped to the welding arms which transmit to the cable the electrical energy required for welding, and also actuate the feed movement of at least one cable end. Usually one arm swings around a pivot. The feed movement and constant pressure at the point of contact during the welding process is maintained by means of an adjustable spring.

The mechanical gear connects the welding arms to the operating lever and to the auxiliary contacts, and serves for starting up and terminating the process. An " off " contact is operated by the swinging arm and is adjustable for any required length of the weld.

Fig. 1 shows a welding set for copper cables of 1.5 sq. mm cross section. It is for use on 50 cycles, 220 V mains. To simplify operation, the melting tube is split and consists of two halves, each with V-shaped cross section. Both parts are held fast in an elastic chuck attached to the fixed arm. Joined together they form a tube of 1.8 mm. inner diameter (corresponding to 1.65 mean diameter of the cable), and of 10 mm. length. The material is " Ardostan," which in a long series of experiments proved to be the most resistive material for the purpose.

The handling of the apparatus is very simple. The cable ends are clamped in the hold-fast, the control lever is depressed and the automatic welding process is begun. When completed all that remains is to loosen the hold-fast and remove the welded cable, which may be used immediately without further work or cleaning.

(*b*) *The Mechanical Properties of the Weld.*

In the example given the weld proper has a diameter of 1.8 mm. and a length of 2 to 4 mm. Adjacent to the weld on both ends is an " intermediate zone " of 2 to 3 mm. length where the cable is not molten but is merely

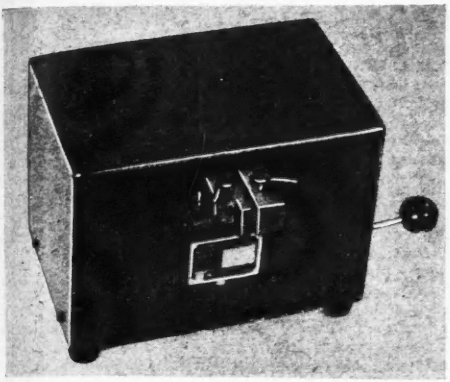


Fig. 1. **Welding set for copper cables of 1.5 sq. mm. cross-section.**

embedded in molten material. The total "stiff length" of the cable created during welding is therefore 6 to 10 mm. (Figs. 4 and 6).

Metallurgical tests of this "intermediate zone" show an intimate diffusion of the copper base with the outer layers of the copper wires (Fig. 3) which proves that this method can be applied also to galvanised wires, in which case the tin coating and the outer layer of the wires form a tin bronze.

As there are no additional chemicals used in this process the weld has the same resistance to corrosion and to wear as the cable itself, and the electrical qualities of the weld are better than those of the cable.

Tensile tests show that the break always occurs a few millimeter from the weld in the short length which had been subject to high temperature, although it was neither part of the weld proper nor of the "intermediate zone." This is due to the fact that the additional tensile strength gained by copper wires when manufactured by cold drawing is subsequently lost by the influence of heat. This decrease of tensile strength remains practically constant for any specific size or type of cable, thanks to the uniformity of the welding process. The divergencies remain within ± 1 per cent. The tensile strength of hard drawn copper cables is, however, well above the corresponding figures for soft copper as the short welding time allows only a partial change in the structure.

The tensile strength of the cable in the example was 40 kg/mm.², the corresponding average figure for the welded cables was 28 kg/mm.², with divergencies within ± 1 per cent., whereas the tensile strength of a similar cable of soft copper is only 23 kg/mm.²

But the decrease of tensile strength is accompanied by an increased elongation, so that if groups of cables are used the welded cable bears a smaller load than the cables with lower elongation and is, therefore, subject to lower stresses.

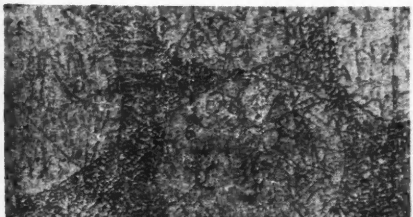
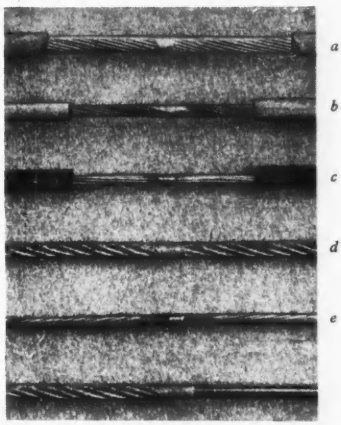


Fig. 2. **Microstructure of the intermediate zone of weld of galvanized copper cable (1.5 sq. mm. cross-sections).**

Fig. 3. **Breaking test of a welded, galvanized copper cable of 1.5 sq. mm. cross-section.**
Break occurred 4 mm. apart from weld. Breaking strength σ_B = 28 kg./sq. mm., Load p_B = 42 kg.

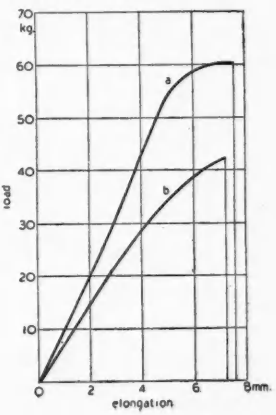


(*a*) Aldrey, 3 sq. mm. cross-section; (*b*) Galvanized copper, 1.5 sq. mm.; (*c*) Steel-copper, 0.9 + 0.38 sq. mm.; (*d*) Bronze, 2.4 sq. mm.; (*e*) Steel, 1.25 sq. mm.; (*f*) Bronze cable (2.4 sq. mm.), welded to bronze wire (3.14 sq. mm.).

Fig. 4. **Different cable welds without subsequent treatment**

3. OTHER APPLICATIONS.

The new welding method is not confined to small copper cables. It can be used for any material suitable for welding, and it is equally suitable for wires as for cables, or for welding cables to wires. It can also be used for welding cables and wires to cable ends in which case it must be specially designed. The tensile strength of two cables or wires of different materials welded



Curve *a*, cable, not welded, Breaking load p_B = 60 kg.
Curve *b*, Welded cable, Breaking load p_B = 42 kg.

Fig. 5. **Load-Elongation diagram of bronze cable of 1.5 sq. mm. cross-section.**

together is about the same as if two ends of the weaker material had been welded.

A regulating transformer and set of interchangeable ceramic tubes are necessary if one welding set is to be used for cables of various diameters. The optimum welding current for non-ferrous materials is approximately 800 amp./sq. mm. and for steel 1,300 amp./sq. mm. If heavier currents are used the welding item becomes too short to allow the material to fill the melting tube homogenously, and the resulting weld is of poor quality. The inner diameter of the melting tube should be about 11 times the outer diameter of the cable, and its length 3 to 6 times (depending on the material and the diameter) the inner diameter of the tube.

For certain purposes it may prove advantageous not to use the split-type but to slip the tube over the cable ends before welding and to break it off after welding thus using a new tube for every subsequent weld. Fig. 6 shows cables welded with a set suitable for cables up to 3 mm. diameter.

The loss of tensile strength of welded steel cables can be largely compensated for by subsequent annealing the parts which had been subject to heat, and which had become brittle by cooling down rapidly when the welding current was switched off.

This annealing can be done advantageously by electric current in the welding set after welding. The current must, of course, be adequately reduced and the necessary switching must be provided.

The method described may be used equally well for welding cables comprising both copper and steel wires. Subsequent annealing is necessary to ensure adequate bending and tensile strengths, and a much heavier welding current must be used.

With regard to the copper wires an average current figure is 5,000 amp./sq. mm. after the relative current distribution between copper and steel wires has been taken into account.

Cables of different material can be welded if they can form an alloy and if their melting points are not too far apart. It is thus possible to weld copper to aluminium and copper to steel. The breaking strength of the welded cables is slightly lower than when both ends are of the same material.

ELECTRIC EQUIPMENT OF PRODUCER GAS VEHICLES

By WERNER KROLL, VDI. (From *ATZ Automobiltechnische Zeitscrift*, Vol. 45, No. 13, 10th July, 1942, pp. 366-369.

1. CHANGE-OVER OF OTTO-CYCLE ENGINES WITH COIL IGNITION.

Starter and Battery.—The compression ratio of coil-ignition engines is about 6 : 1, but by making use of the higher anti-knocking property of gas-air mixtures, the ratio may be increased to 8 : 1. More power is, therefore, required for starting, and to provide this several methods may be adopted.

(*a*) Fitting a larger battery with the maximum A.H. capacity to suit the existing starter.

(*b*) Replacing the 6V or the 12V starter by a special starter of the same size, designed for double the voltage. A second 6V or 12V battery should be provided and connected in series during starting by a special switch.

(*c*) Fitting a larger starter, or two : one at each side of the engine. This will, however, seldom be possible.

(*d*) General measures, which assist the battery at lower temperatures, are : turning the engine over by hand before starting, warming up the engine by external means, and warming up the battery to about 20 deg. C., to increase the battery power. (See Fig. 1.)

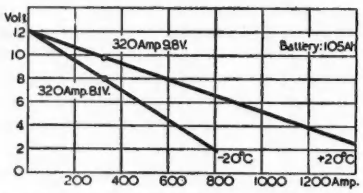


Fig. 1. **Decrease in battery power at low temperatures.**

Dynamo.—If a larger starter is fitted, the existing dynamo may be insufficient, more so if an electric fan of 100–200 W is used during starting of the gas-producer. A higher powered dynamo must, therefore, be fitted, preferably with some form of voltage-regulation, if this does not already exist.

Coil and Distributor.—As a result of increased compression ratio, ignition voltages are higher and the existing coil must be replaced by a higher power coil. The distributor must be arranged to give more advance, because a gas-air mixture has slower combustion speed than petrol-air mixture. (Fig. 2.) It is recommended that graphs be made for each type of engine.

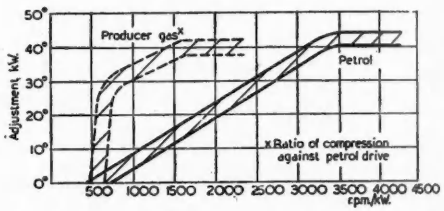
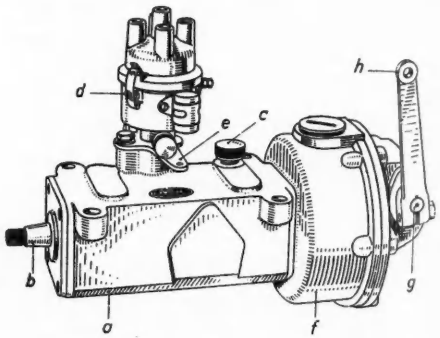


Fig. 2. **Modification of ignition regulator characteristics for gas engines as compared with petrol engines.**

Sparking Plugs.—Longer heat action and chemical attacks cause the plug points to burn away more rapidly. Plugs of higher heat transmission ratio should, therefore, be fitted and, to avoid excessive voltages, the gap is usually reduced to 0.4–0.5 mm., but, if the compression ratio is not too high, the normal gaps may be retained. It is of particular importance to readjust the gaps as soon as this becomes necessary, and for cleaning dirty plugs, sand-blasting has proved to be an efficient method.



(*a*) Gearbox ; (*b*) Driving shaft ; (*c*) Oil dip stick ; (*d*) Distributor ; (*e*) Adjusting lever (for ignition) ; (*f*) Centrifugal governor ; (*g*) Control shaft ; (*h*) Control lever (coupled to throttle).

Fig. 3. **Bosch change-over ignition set.**

Cooling bad when using an intermediate plug ; remedied by modification of the cylinder head.

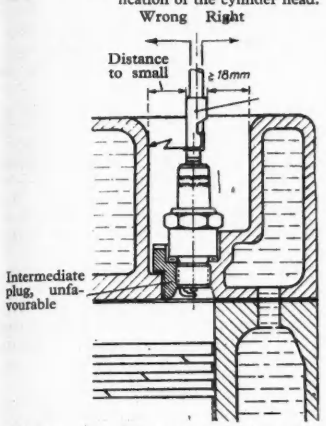


Fig. 4. **Cylinder head of an adapted diesel engine.**
Left : Cylinder head unchanged ; Right : Modified head.

H.T. Leads.—The voltage being usually higher, well-insulated H.T. cables with lacquered sheathing are necessary. Ordinary rubber-insulated cables soon become brittle, due to chemical attack by ozone, generated by the creep of luminescent discharges. The increased voltages also increase capacity-coupling between adjacent leads, and may even build up electrical charges in a cylinder during the suction or compression stroke. The cables must be arranged and enclosed in a metal tube well earthed to the body of the engine. It would be better still to lay the cables in metal tubes individually or to use metal-sheathed cables. An efficient means of relieving difficulties when starting is by fitting vibrator-ignition.

2. CHANGE-OVER OF OTTO-CYCLE ENGINES WITH MAGNETO IGNITION.

Magnetos.—Magnetos of normal type are insufficient because of their low tension and a high-powered type of magneto should therefore be fitted.

Starter, Battery, Dynamo.—These parts should be treated in the same way as stated for coil-ignition engines.

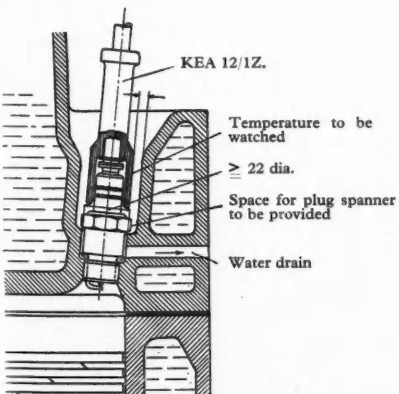


Fig. 5. **Cylinder head with deep recess.**

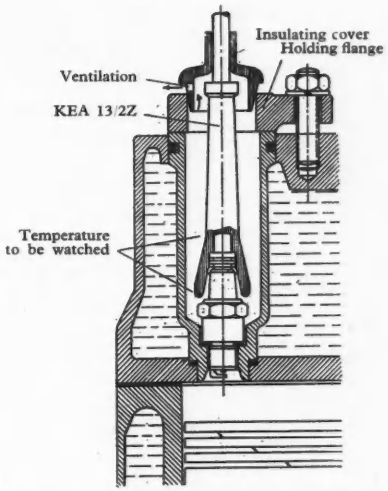


Fig. 6. **Plugs fitted in place of a pre-combustion chamber.**

3. CHANGE-OVER OF DIESEL ENGINES TO SPARK IGNITION ENGINES.

Ignition Coil and Distributor.—A high-powered coil should be used. For the distributor, Bosch have designed a change-over ignition set which fits in the place of the injection pump. It contains the necessary angular drive and a centrifugal speed governor. A petrol pump may be connected to the change-over set, if required. This governor is sufficiently strong to move the mixer throttle, which may easily stick, because of tar in the gas. Simple distributors without a governor should not be fitted. Electrical governors, which cut out ignition at a given maximum engine speed, are also dangerous, because during the time the ignition is switched off, gas-air mixture can accumulate in the exhaust silencer and may explode when the ignition is switched on.

Sparking Plugs.—Fitting sparking plugs in the place of injection nozzles is extremely difficult. To ensure reliable working, the plugs must be correctly placed ; good cooling must be provided ; sufficient space kept between H.T. parts of the plug and the engine-body, and the electrodes must reach into the combustion space. If any of these conditions cannot be fulfilled, when using the existing C.I. cylinder head, it is better

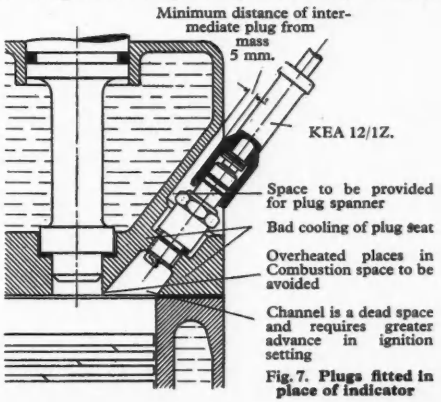

Minimum distance of intermediate plug from mass 5 mm.
KEA 12/1Z.
Space to be provided for plug spanner
Bad cooling of plug seat
Overheated places in Combustion space to be avoided
Channel is a dead space and requires greater advance in ignition setting

Fig. 7. Plugs fitted in place of indicator

to fit a new head altogether. The left-hand side of Fig. 4 shows a cylinder head adapted for producer-gas, and the right-hand side a correctly designed new head. The 18 mm. distance shown is a minimum. Fig. 5 shows the precautions which should be taken when plugs are to be fitted in a deep recess. Plugs may often be fitted in place of a pre-combustion chamber (Fig. 6). They must reach into the combustion space, otherwise exhaust gases will accumulate in the channels shown in Fig. 7, and make ignition more difficult. Another disadvantage of hidden plug points is the long path from the spark to the cylinder centre. This demands a greater advance and, as a result, increases the temperature. The sharp edge seen in Fig. 7, which is unintentionally provided, leads to slow ignitions when using producer-gas, and must be eliminated. Generally, 18 mm. plugs are best for gas-engines, as they are less prone to become dirty and have a longer life. 14 mm. plugs must often be fitted, because of the small space available, but the smaller plugs are not recommended.

Dynamo.—The dynamo need not be replaced, as diesel engines are usually fitted with dynamos of large enough size.

Starter and Battery.—As the compression ratio is always lowered and the starting power required for turning over the engine is reduced, the starter and battery may remain unchanged.

4. CHANGE-OVER OF DIESEL ENGINES TO THE DIESEL-GAS SYSTEM.

In this system the engine is run on an air-gas mixture, ignition taking place at the end of the compression stroke, by injected oil. Depending on engine design, 15–25 per cent. of the original oil consumption is needed for ignition. The electrical equipment remains unaltered.

The system is used chiefly to adapt existing engines speedily and at low cost. Despite its advantages, a large-scale adaptation of diesel vehicles is out of the question at present, because the quantity of oil required for the combined type of engine is needed for other purposes.

PRESSURE COMBUSTION ACCORDING TO THE VELOX PRINCIPLE.

By Dr.h.c. W. G. NOACK VDI. (From *Zeitschrift des VDI*, Vol. 87, No. 35-36, September 4th, 1943, pp. 547-555.)

BOTH the recuperators and regenerators which heat the hot blast required in blast furnace operation can be built on the Velox principle. In recuperators of this type (Fig. 1), the products of combustion are mixed with recirculated waste gas in order to keep the combustion chamber temperature within 1,100-1,200 deg. C. Particular attention must be paid to tube wall temperatures, as on their elevation both the obtainable hot blast temperature and the service life of the heater tubing are dependent. In this respect the Velox heater possesses advantages over the atmospheric type, insofar as the greater permissible pressure drop permits the grading of gas and blast velocity and of the respective heat transfer, in such a manner as to obtain a small temperature margin between tube wall and blast at the hot end and a correspondingly higher margin at the cold end. Furthermore, products of combustion and blast may be at identical pressure, thus relieving the tubing of pressure differences, and permitting employment of a tube material combining highest scaling resistance with moderate high temperature strength. If scaling resistance alone were to be considered, Velox blast heaters could be built at the present time for the highest blast temperature called for in blast furnace operation. Unfortunately, however, scaling resistance depends upon the cleanliness of the gas, a condition which is not met by blast furnace gas. It must, therefore, be feared that at high temperatures, the protective oxide skin of the tube will be destroyed by intercrystalline corrosion induced by the chemical attack of the dust and slag

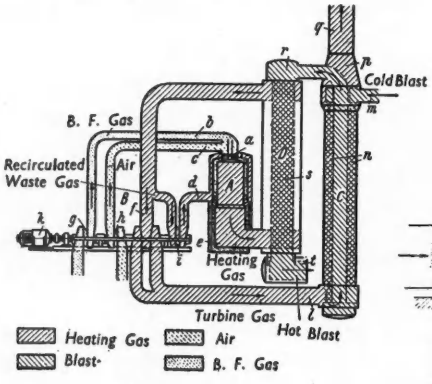


Fig. 1

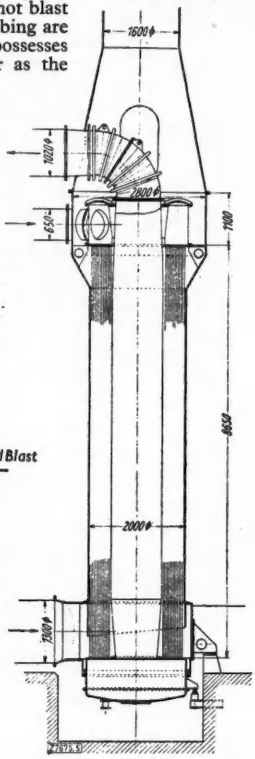


Fig. 2

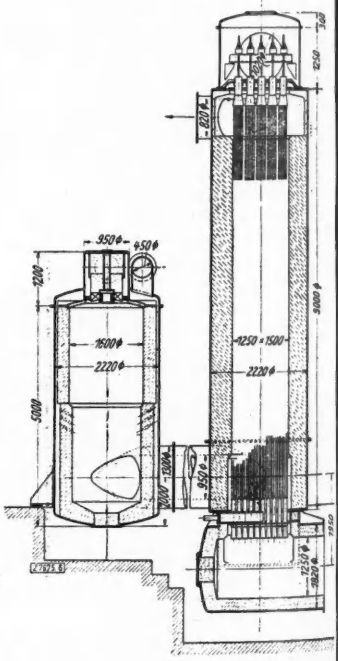


Fig. 3

contents of the blast furnace gas upon the tube surface. For the time being, tube metal temperatures must therefore be kept below 850 deg. C., corresponding to a maximum hot blast temperature of 750 deg. C. continuously and 800 deg. C. intermittently.

Heating is accomplished in two stages, with the gas turbine inserted between the stages. The first heating stage comprises a pre-heater receiving the waste gas of the turbine as heating medium. As temperatures in this stage lie below 400 deg. C., ordinary steel tubing can be used. A typical first stage heater is shown in Fig. 2, while the second stage heater is shown in Figs. 3 and 4. Referring to the latter illustration, each tube bundle is composed of 16 tubes which at their cold ends are welded into the head (a), their lower hot ends being formed by the nipples (b), each nipple combining four tube ends. The tube bundles are seen to be suspended from sturdy springs (c) which pull the end rings (d) against the bottom tube sheet (e) affixed to the I-beams (i), which also support another bottom sheet (h). Into the false bottom enclosed by the sheets (e) and (i), cold air is blown for cooling purposes. At the top, the inlet ends (a) of the tubes are sealed by the bushes (g) arranged in the air-cooled beams (f). Each tubular element can be withdrawn individually by means of block and tackle arranged for this purpose. The size of the heater must naturally depend upon the permissible pressure drop. If this may be increased to 1,500 mm. w.g., the heating surfaces required will amount to only ½ to 1/5th of those installed in the usual heater types.

Particularly advantageous conditions for the use of the Velox blast heater exist in cases where a common blower set is used for compressing both combustion air and blast, while both blast and heating are heated by the combustion gases of the same combustion chamber. A layout of this type, as shown in Fig. 5, secures especially favourable operating conditions for the blower, since the increase in blower capacity called for by the combined delivery of combustion air, cooling air, and blast leads to a rise in blower efficiency. At the same time, the blast itself represents only a fraction of the total output of the blower, so that fluctuations in blast requirements and blast pressure have little effect upon the operating conditions of the blower. This stability of operating

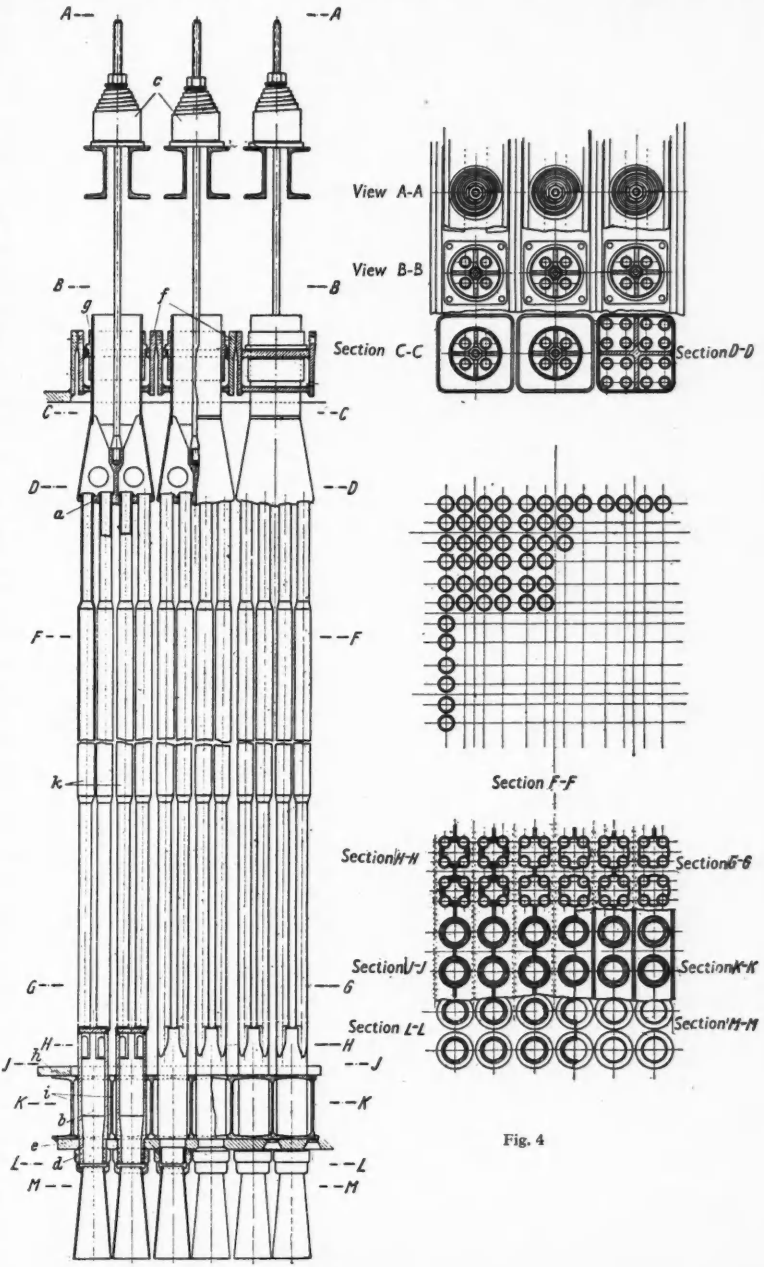


Fig. 4

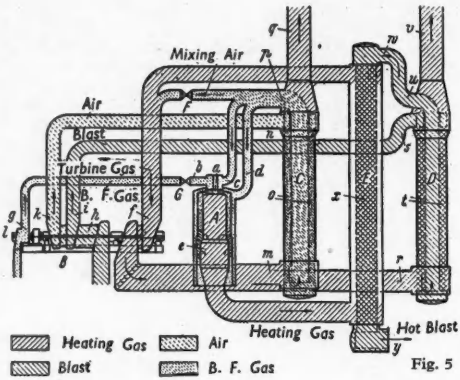


Fig. 5

conditions makes it possible to employ a blower of the axial type which, compared to radial or centrifugal blower types, shows superior efficiency at normal load, but loses this advantage with declining loads.

Furthermore, as the products of combustion leaving the combustion chamber are cooled in the second blast heater stage before they are admitted to the gas turbine, there is no need for the employment of the large amount of cooling air usually required. Also, since the same heating gas serves both blast heater stages, waste gas loss occurs only once. As shown in Fig. 5, the turbine is placed between the two stages of the blast heater, the waste gases discharged from the turbine serving to heat the first stage heater. The pressure of heating gas is usually chosen 1.5 times that of the blast, the latter being taken from an intermediate blower stage. As Fig. 5 shows, separate heaters are used for blast and combustion air, each of them receiving one-half of the products of combustion.

WOOD USED AS MATERIAL FOR CHEMICAL APPARATUS

By DR.-ING. H. POETTER, Mako-Werke, Erfurt. (From *Die Chemische Technik*, Vol. 16, No. 9, May, 1943, pp. 81-82)

WOOD is a very suitable material for building chemical apparatus and has been found to be an excellent substitute for scarce metals such as alloyed steels. Its use for building chemical apparatus, however, presumes an intimate knowledge of its behaviour to chemical reagents.

It has high resistance qualities against light, air, and humidity, but its stability is somewhat limited at higher temperatures, although at a temperature of 125 deg. C. it can be used satisfactorily.

Wood is completely dis-integrated by concentrated mineral acids, such as sulphuric, hydrochloric and phosphoric acid ; the resistance to dilute mineral acids is fairly high and exceeds all expectations, but it cannot naturally be compared with the resistance of alloyed steels. Sulphurous acid gradually dissolves the lignin leaving the cellulose untouched, while sulphuric and hydrochloric acid attack the cellulose contained in the wood.

Alkalies react only with the surface layers of the wood. Thus, ammonia in water solution or as gas has only a noticeable effect at high pressures and temperatures. For salts the same applies as mentioned for acids and alkalies.

Wood is less resistant to oxidizing agents, such as nitric acid and chromic acid, while oxidation in an alkaline medium takes places at a much slower rate, extremely large quantities of oxidizing agents are required. The same remarks apply to potassium-permanganate and hydrogen-peroxide.

Organic acids are similar in their behaviour to mineral acids, with the exception of acetic acid which does not attack the cellulose part of the wood. The destructive effect of all chemical reagents is considerably decreased by keeping the pressure and temperature as low as possible.

The use of wood in chemical apparatus requires an exact knowledge of the material itself. It must be rich in resin, slowly grown, about 70-80 years old, and without knots—as far as possible.

The opinion that only pitch-pine and similar wood such as jugo-pine are suitable for chemical apparatus has been proved wrong in practice. Northern pine and German larch have been proved satisfactory. A good high mountain larch is always to be preferred to a second class pitch-pine.

Based on this knowledge chemical apparatus of the largest dimensions are now made of wood. These comprise mordant baths, stirrers, dissolving vessels, neutralization vessels up to 200 cu. m. capacity, suction strainers—even vacuum filters—evaporators, cooling towers, wooden pipe lines and ventilators.

Two years' experience in this direction gave results which exceeded all expectations. It showed that also from the economical point of view, wood is a very suitable substitute material. Two examples are given below to illustrate this.

1. In the explosives' industry an aluminium apparatus with a wall thickness of 10 mm. working at relatively low temperatures had been used ; this was replaced by a wooden apparatus of 75 mm. wall thickness. The durability of the apparatus should be roughly two and a half times that of its aluminium equivalent, but the cost of the wooden apparatus was approximately 50 per cent. less.

2. In the same industry an apparatus of high quality alloy steel with 5 mm. wall thickness was used formerly and is now replaced by a wooden apparatus of 75 mm. wall thickness. The cost was only 2/3 that of the steel, but the durability was only 1/2 that of the steel apparatus.

In this case the lower durability is not balanced by the smaller cost. This was compensated for by smaller heat losses, reduced fuel consumption, and furthermore by the far more favourable repair and maintenance conditions.

The two examples mentioned covered a single order and represented a saving of 224 tons of aluminium, 45 tons of chromium and 23 tons of nickel.

ZERO RESISTANCE CIRCUIT FOR PHOTO-CELL PHOTOMETRY.

(From *Technical News Bulletin*, No. 323, March, 1944, pp. 18-19).

" ZERO-RESISTANCE " circuits are widely used for accurate photometric measurements with barrier-layer photocells. T. H. Projector, M. K. Laufer, and C. A. Douglas have made several improvements in a circuit of this type, using a Kohlrausch slide-wire. A standard cell and control rheostats have been added to the circuit in such a way that the e.m.f. of the standard cell can be balanced against the IR drop across a resistor in the slide-wire circuit. By this means the slide-wire current is kept constant. A " damping selector " is used with the galvanometer in order to provide a range of operation from slightly underdamped to very much overdamped. Under adverse conditions, overdamping makes the galvanometer less responsive to disturbing transients and thus facilitates setting the slide-wire for null deflection of the galvanometer.

A Leeds and Northrup K-2 potentiometer is used in a " zero-resistance " circuit. A simple selector switch makes it possible to change over conveniently from photometric to ordinary potentiometric measurements.

ELECTRONIC TUBES, HOW THEY WORK.

(From *General Electric Review*, Vol. 46, No. 11, November, 1943, pp. 612-617).

An electronic tube has four basic parts: (1) the cathode; (2) the anode; (3) a glass or metal envelope; (4) terminals for connecting the tube into the circuit. Some tubes have a fifth part, the grid, which controls the current passing through the tube. Many tubes have a separate heating coil to bring the cathode to the temperature required to make it emit electrons.

In the following description of these basic tube parts, all letters A, B, C, etc., refer to the chart on page 154.

The Heater (*A*). There are two methods of heating: (1) to build the cathode in the form of a filament which is heated by its own resistance when a current is passed through it; and (2) to use a separate heating coil inside heat conserving shields. This heating coil is called the heater.

The Cathode (*B*). In the electronic tube the circuit is broken: the cathode is one end of the broken conductor and the anode is the other. To re-establish a flow of current through the circuit, electrons must flow across the gap between the cathode and the anode. Electrons cannot get across this gap without assistance.

The Anode (*C*). The anode attracts the billions of electrons that are "boiled" off the surface of the cathode, thus establishing a flow of current through the tube. The anode is frequently referred to as the plate, and the current is called the plate current. At the proper operating temperature the anode cannot give off electrons.

In an a.c. circuit, the anode is positive during half of each cycle, during which current current flows through the tube. During the other half of the cycle current cannot flow through the tube. It is this characteristic of an electronic tube which enables it to act as a rectifier.

The Grid (*D*). The grid, situated between the cathode and the anode, is the controlling electrode of the tube. Some tubes have several grids to permit greater range of control.

The grid (D) controls only the point in the cycle at which electrons will start to flow to the anode. For example, if the tube is being used to control a resistance-welding circuit in which the welding time is required to last only five one-thousandths of a second, the grid can be made to hold the current off until the last five one-thousandths of a second of the positive side of the cycle.

The grid will not permit electrons to reach the anode as long as a certain negative voltage value is maintained on the grid. When this voltage becomes less negative, the grid works like a trigger, suddenly permitting the full flow of electrons to start toward the anode. Once the flow of electrons has started, the grid in any gas-filled tube loses control over the flow until the next cycle; that is until the anode has become negative and stopped the flow of electrons. In a high-vacuum tube, the grid never loses control over the flow of electrons. Instead, the flow is at all times proportional to the voltage applied to the grid, and any change in grid voltage causes a proportional change in the amount of current flowing through the tube. The power required by the grid, in both the gas-filled and high-vacuum tubes, is very small. Therefore, a relatively small amount of power in the grid circuit can precisely control a much larger amount of power in the plate circuit.

The Terminals. The terminals on the tube provide for connection of the cathode and anode into the main circuit, and connection of the heater and grid to their respective supply circuits.

The Envelope. The chief function of the envelope is to maintain the vacuum in the tube. In some tubes it also has an important function in dissipating the heat created in the tube so as to keep it at a proper working temperature. In making gas-filled tubes, the air is first pumped out until a high vacuum is created. Then, before the tube is sealed, a tiny quantity of argon, neon, mercury, or some similar substance is inserted. This gas or mercury is chemically inert, which means that it will not combine with any of the metals used in the tube.

The major distinctions between gas-filled and high-vacuum tubes are these: (1) High-vacuum tubes work like a throttle in that they provide continuous and uniform control of the amount of current flowing through the tube: (2) Gas-filled tubes work like a switch. They are capable of accurately and instantaneously closing a heavy-current circuit; (3) Gas-filled tubes will carry heavy current much more efficiently than high-vacuum tubes of the same size; (4) High-vacuum tubes can be used as distortionless amplifiers; gas-filled tubes cannot.

The effect of the billions of electrons passing between the cathode and anode is to create a strongly negative electric field. This field is known as space-charge and is the basic reason for the inability of high-vacuum tubes to carry as large a current for their size as gas-filled tubes.

Electrons emitted by the cathode travel with tremendous velocity and when one of them collides with a gas molecule the impact knocks one or more electrons out of the molecule. These molecules, which have lost some of their electrons and are now positive, are called ions. The process is called ionization and is responsible for the greater current-carrying capacity of gas-filled tubes.

Assume that a typical current—60-cycle, 110-volt alternating current—is being used in the cathode-to-anode circuit of the tube. Such a current changes direction 120 times a second, twice each cycle. What is shown in (H) takes place in 3/120 of a second.

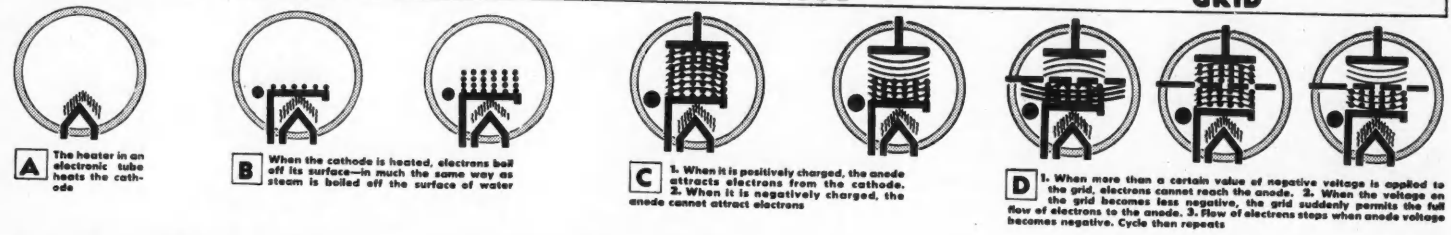
Because electrons can leave the cathode but cannot leave the anode, electronic tubes permit current to flow in only one direction and only during the positive portion of each cycle. The negative portion of each cycle is eliminated by the tube. The output of the tube is, therefore, a pulsating direct current. The pulsations can be removed by filters. Diagram (H) illustrates the output of a tube having no control grid. Since it is utilizing only half of the current wave (one side of the cycle only), it is known as a half-wave rectifier. If two tubes are connected so that the cathode of one is connected to the anode of the other, thus conducting both halves of the cycle, the output will be an alternating current which can be controlled cycle by cycle. This type of circuit is used for resistance welding.

Assume now that we replace this rectifier tube with a high-vacuum tube containing a grid. Assume also that the grid is set to reduce the flow of current through the tube by one half. Diagram (I) shows the output wave of this tube. (Dotted line represents the maximum output of the tube; solid area is the output permitted by the grid).

The diagram shows that the controlled-output wave has the same shape as the full output wave. This means that the tube has exerted a continuous and uniform control of the current during the entire period of flow. This control can be used for amplification. In a gas-filled tube in which the grid is set to reduce the flow of current through the tube by one half (K), the output diagram has not the same shape as the uncontrolled output wave. It shows that the current did not flow at all during the first quarter of a cycle. At this point, the full flow of current suddenly came on and remained on for a quarter of a cycle. During the negative half cycle the current remained off, as it did in the vacuum tube.

The diagrams illustrate the basic differences between high-vacuum and gas-filled tubes. The high-vacuum tube works like a rheostat, the gas-filled tube works like a switch.

ELECTRONIC TUBES . . . how they work



A The heater in an electronic tube heats the cathode

B When the cathode is heated, electrons boil off its surface—in much the same way as steam is boiled off the surface of water

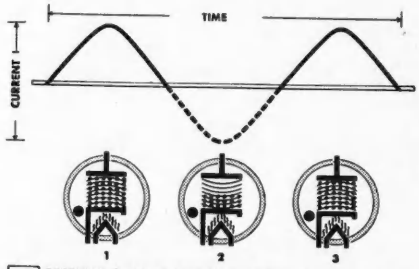
C 1. When it is positively charged, the anode attracts electrons from the cathode. 2. When it is negatively charged, the anode cannot attract electrons

D 1. When more than a certain value of negative voltage is applied to the grid, electrons cannot reach the anode. 2. When the voltage on the grid becomes less negative, the grid suddenly permits the full flow of electrons to the anode. 3. Flow of electrons stops when anode voltage becomes negative. Cycle then repeats

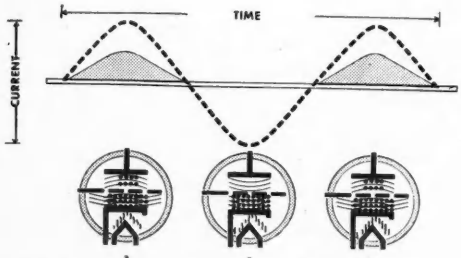
E Symbol for a gas-filled tube

F Symbol for a high-vacuum tube

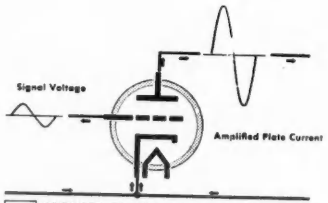
G IONIZATION 1. An electron (negative charge) is knocked out of neutral gas molecule by another electron from the cathode. 2. The gas molecule is now positive. 3. Positive molecules are called ions. 4. Ions tend to neutralize most of the space charge (negative electric field caused by electrons passing from cathode to anode), permitting more electrons to reach the anode. Ionization is responsible for the greater current-carrying capacity of gas-filled tubes



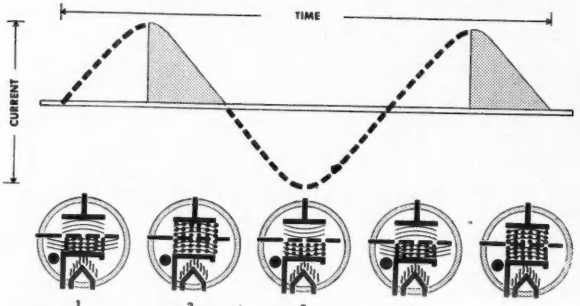
H RECTIFIER 1. Anode is positive and attracts electrons. 2. Anode is negative and does not attract electrons. 3. Cycle repeats—each step lasts 1/120 second. Note that negative portion of each cycle (dotted line) is eliminated



I GRID ACTION of a high-vacuum tube. 1. Grid is cancelling part of effect of anode; only part of electrons are reaching anode. 2. Anode is negative and does not attract electrons (no current flows). 3. Cycle repeats. Note continuous and uniform control of current flowing through the tube (solid area)



J AMPLIFIER—The plate-current wave takes exactly the same shape as the signal voltage. Since the plate current is many times larger than the grid current, this results in amplification



K GRID ACTION of a gas-filled tube. 1. Grid prevents flow of electrons—completely cancels effect of anode. 2. When less than a minimum voltage is applied to grid, full flow of electrons is suddenly released (solid area). 3. Anode is negative and does not attract electrons. 4-5. Cycle repeats

TECHNICAL NEWS

Announcements in this section include News relating to British Firms, and on Equipment produced by British Manufacturers.

Available literature may be secured by addressing a request to the advertising department of "The Engineers' Digest," or by writing direct to the manufacturer and mentioning "The Engineers' Digest" as a source.

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- ✦ BOOKS AND CATALOGUES
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- ✦ PERSONAL
- ✧ NOTES

INCANDESCENT THERMOBLOC RECUPERATORS AND HEAT EXCHANGERS.

MUCH of the sensible heat in fuel fired furnaces is lost in the waste gases, and the recovery of a proportion of this heat is desirable from the standpoint of economy and also because heat recovered can be usefully applied to improve combustion conditions and facilitate atmosphere control in the majority of furnaces. Waste heat recovery has been standard practice for many years, and the majority of large industrial furnace installations are either on the regenerative principle or on the recuperative principle.

The regenerative principle is successfully applied to large high temperature furnaces, either for the melting of metals or for high temperature reheating for forging, whereas the recuperative principle is more often adopted primarily because it is easier to obtain precise temperature control than with the reversing cycle implicit to regenerative furnaces.

Developments of alloy steels capable of withstanding relatively high temperatures, has facilitated the use of metallic recuperators and heat exchangers in place of brickwork recuperators, and for many applications a metallic recuperator has proved highly satisfactory because of its accessibility, low thermal capacity, and gas tight characteristics which enables the preheating not only of the combustion air or the gaseous fuel, but also of mixtures of gas and air.

The use of metallic recuperators has been limited owing to the high capital cost of such heat exchangers manufactured in special alloys, such as nichrome, to withstand the relatively high temperature conditions met with in practice, and it has long been desirable to develop an efficient type of metallic recuperator of small bulk and low first cost enabling it to be applied to the widest possible range of industrial heating problems.

The Patent Incandescent Thermobloc manufactured by the Incandescent Heat Co., Ltd., Birmingham, and described below covers this requirement. This unit can deal with waste gases at temperatures in excess of 1,000 deg. C., without the use of expensive heat resisting alloys in its construction, and the rate of heat exchange per sq. ft. of surface area is so high that the body of the Thermobloc can be kept at a relatively low temperature.

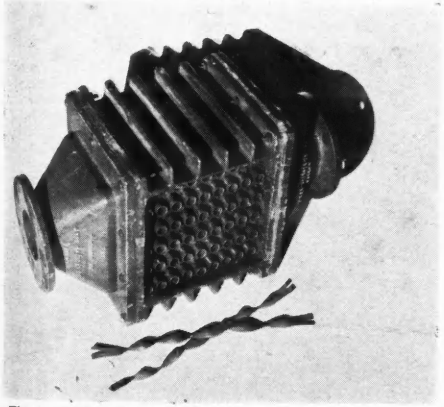

Fig. 1. **"Thermobloc" Recuperator with Corebusters.**

The overall dimensions of the No. 1 Thermobloc are 9 × 9 × 12 ins., the heat exchanging surface 5 sq. ft., and 9 × 12 × 15 ins., the heat exchanging surface 10 sq. ft. for the No. 2 Thermobloc.

Spiral corebusters in the mixture passages ensure an intimate premix of the gas and air and higher efficiency combustion with minimum stratification.

The Patent Incandescent Thermobloc (Fig. 1) consists essentially of two banks of small diameter tubes. One bank is disposed vertically in the cast iron matrix, and the other is horizontally interlaced between the vertical bank. Usually the vertical bank is used for the passage of waste gases, whilst the horizontal bank is used for the medium to be preheated.

Waste gases are thus split up into a number of streams ensuring a very large surface area for the release of sensible heat into the horizontal bank of tubes. In similar manner the gas or air entering the horizontal bank of tubes is split up for the absorption of heat

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Fig. 2. **Application of Recuperators to a batch of furnaces for hardening high-speed steel.**

from the waste gases. Heat absorption is further intensified by the provision of spiral corebusters designed to break up the relatively cold core in the centre of each tube and throw it outward towards the periphery by centrifugal action.

In addition the particular arrangement of tube banks adopted ensures the highest possible heat exchanging surface per unit of volume of the Thermobloc, and it is interesting to note that the finished Thermobloc is no less than 35 per cent. hollow. The units can be used singly or in banks according to requirements, and generally their use permits an economy of at least 75 per cent. of the space required for a metallic recuperator of given capacity, as compared to heat exchangers of conventional design.

The Thermobloc Preheater or heat exchanger was originally designed to meet conditions whereby it was required to fire existing Town Gas Industrial Furnaces on low C.V. gas, such as clean producer gas. This problem presented few difficulties for working temperatures up to the order of 1,000 deg. C., but owing to the low flame temperature of this gas it was obviously necessary to provide some form of preheat if producer gas was to be successfully used on high temperature furnaces up to 1,500 deg. C., or more. Fig. 2 shows the application to a batch of high-speed tool-hardening furnaces heated by clean producer gas, in which temperatures up to 1,500 deg. C. can be readily obtained and maintained, thereby permitting the use of the furnace at somewhat lower temperatures of say 1,350 deg. C., with the heavily reducing atmosphere necessary for the critical treatment of delicate tools and cutters. It is interesting to note that a small high-speed-steel furnace shows an economy of 25 per cent. of the gas consumption when using a Thermobloc for preheating the air, as compared to the operation of the same furnace without preheat.

Thermoblocs may often be applied to existing furnace installations, with a resultant economy of 20 per cent. at least, quite apart from improved combustion conditions, accurate atmosphere control, and consequent improvement in the protection of the metal under treatment. It is generally possible to recover the capital cost involved in the installation of a preheater, in a period of 12 to 18 months by the direct saving in fuel.

In certain cases, such as high temperature open flame work with low C.V. gases, Thermoblocs can be successfully applied as independent preheaters.

Although the Themobloc was primarly designed as a heat exchanger or recuperator, capable of dealing with relatively high temperatures, the same design has proved highly successful as applied to heat exchangers for lower temperatures wherein the media to be dealt with are usually in liquid or fluid form. Thermoblocs can be built up readily into banks to form industrial oil coolers, and further can be successfully applied to a wide range of problems involving the cooling or heating of liquids and gases to relatively low temperatures. They can also be manufactured in special metals for handling food products, or chemicals.

Hiduminium Applications Ltd.—Light metals will play an indispensable part in solving the mechanical and structural problems of the future. A new company has been formed by HIGH DUTY ALLOYS LTD., REYNOLDS TUBE CO. LTD. and REYNOLDS ROLLING MILLS LTD., to collaborate with designers and constructors in any industry to secure the best use of Hiduminium Alloys.

Further information will be sent on request to Hiduminium Applications Ltd., Farnham Road, Slough, Bucks.

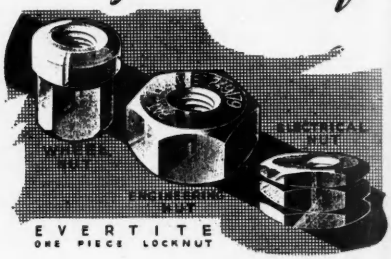
PERSONAL.

Mr. J. L. Munn has been appointed Works Manager of the Austin Motor Company Ltd. Mr. Munn is Chairman of the Midlands Centre of the Institution of Production Engineers.

Dr. Andrew McCance has been elected deputy Chairman and joint Managing Director of Colvilles Ltd., Glasgow.

Mr. M. Burningham has been elected deputy Chairman of Keith Blackman Ltd.

Can a FORCE act against itself!



We will not argue about the feasibility of it—in a general sense. All we can say is, that in the case of the EVERTITE Locknut, increasing vibration defeats its own purpose by serving to increase the grip of the EVERTITE on the bolt threads. Put another way—EVERTITE utilises vibration as an ally for nut tightness.

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